VOL. XXXIV. — No. 10. OCTOBER 1957.

Monthly Bulletin of the International Railway Congress Association

(English Edition)

RISS

"Limpet" Asbestos Mattresses pay their way in Locomotive Boiler insulation...

Tests show that in the first year "Limpet" Asbestos Mattress insulation can pay for itself in fuel saved.

Moreover, these well-fitting envelopes of asbestos cloth filled with pure asbestos fibres are robust enough to last at least as long as the boiler.

"Limpet" Asbestos Mattresses are accurately tailored to fit snugly to the boiler plates, preventing particularly those heat losses caused by forced convection when the locomotive is in motion. They are made to stringent specifications, are light in weight, and their fixing and maintenance costs are negligible.

LIMPET BRAND
TRADE MARK REGISTERED

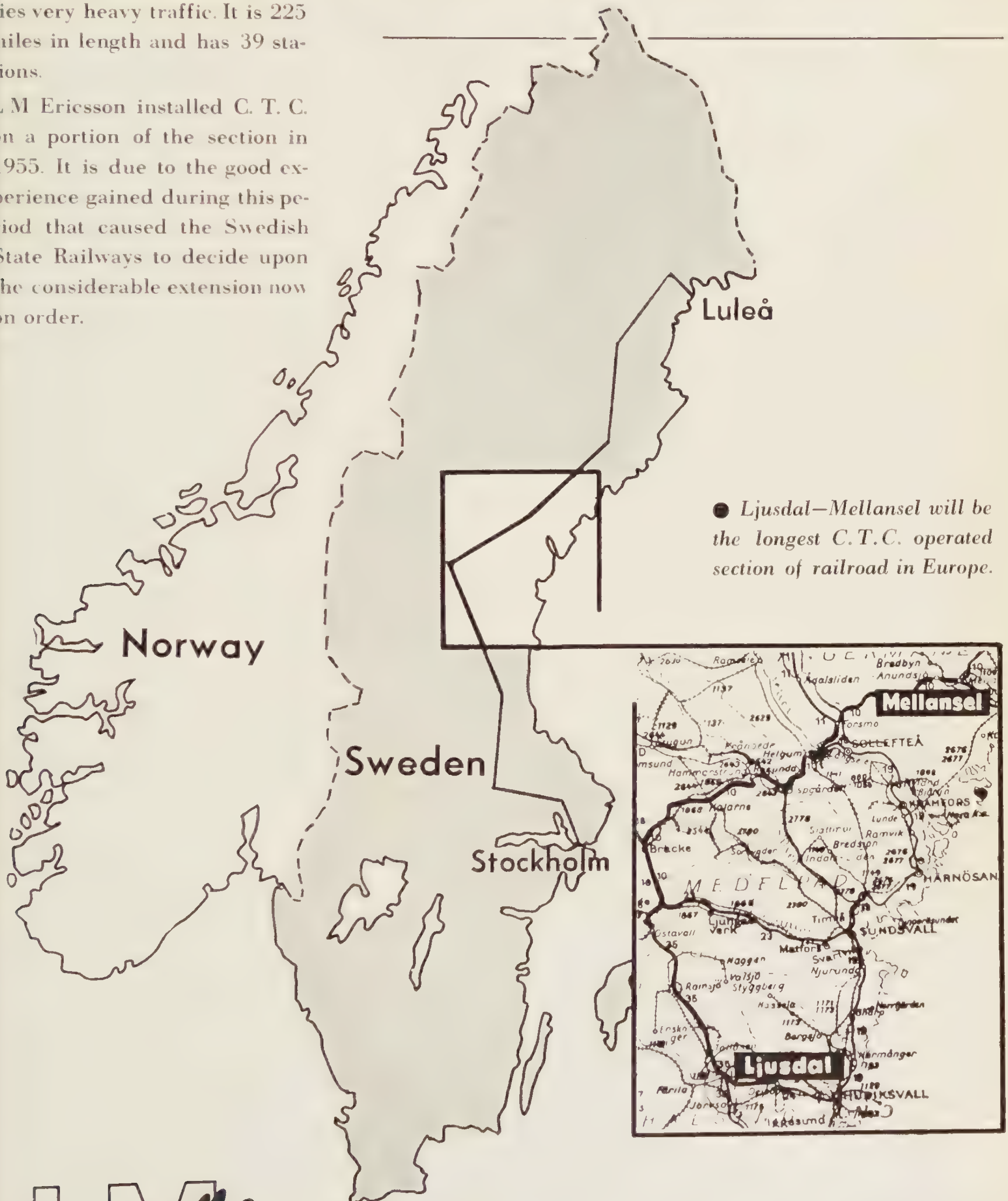
J. W. ROBERTS LTD · ARMLEY · LEEDS 12 . ENGLAND

A MEMBER OF THE TURNER & NEWALL ORGANISATION

The Swedish State Railways have ordered LM Ericsson C.T.C. equipment for the Ljusdal—Mellansel section of the northern main line. The section is single track and carries very heavy traffic. It is 225 miles in length and has 39 stations.

L M Ericsson installed C. T. C. on a portion of the section in 1955. It is due to the good experience gained during this period that caused the Swedish State Railways to decide upon the considerable extension now on order.

Swedish State Railways install C.T.C. on 225 miles of line



● *Ljusdal—Mellansel will be the longest C.T.C. operated section of railroad in Europe.*

LM ERICSSONS SIGNALAKTIEBOLAG

Lövholmsvägen 93 Stockholm Sv Sweden


TR 216
FIAT 077B RAILCAR IN OPERATION ON GREEK RAILWAYS
RAILWAY JOURNAL AXLE-BOXES
RIV
OFFICINE DI VILLAR PEROSA S. p. A. - TORINO

TELOC with electrical remote drive
Hasler Berne
TELEPHONE EQUIPMENT AND

Alphabetical Index of Advertisers

MONTHLY BULLETIN

OF THE

INTERNATIONAL RAILWAY CONGRESS ASSOCIATION

(ENGLISH EDITION)

PUBLISHING and EDITORIAL OFFICES: 19, RUE DU BEAU-SITE, BRUSSELS

Price of this single copy : 80 Belgian Francs (not including postage).

Yearly subscription for 1957 { Belgium . . . . 700 Belgian Francs
Universal Postal Union . . . . 800 Belgian Francs

Subscriptions and orders for single copies to be addressed to the General Secretary, International Railway Congress Association, 19, rue du Beau-Site, Brussels (Belgium).

CONTENTS OF THE NUMBER FOR OCTOBER 1957.

CONTENTS *(continued).*		Page.

LIBRARY

OF THE

Permanent Commission of the International Railway Congress Association

READING ROOM : 19, rue du Beau-Site, Brussels.

Works in connection with railway matters, which are presented to the Permanent Commission are mentioned in the « Bulletin ». They are filed and placed in the library. If the Executive Committee deems it advisable they are made the subject of a special notice. Books and publications placed in the reading room may be consulted by any person in possession of an introduction delivered by a member of the Association.

Books, etc., may not be taken away except by special permission of the Executive Committee.

The Permanent Commission of the Association is not responsible for the opinions expressed in the articles published in the « Bulletin ».

An edition in French is also published.

Vol. XXXIV. — No. 10. October 1957.

BULLETIN
OF THE
INTERNATIONAL RAILWAY CONGRESS
ASSOCIATION
(ENGLISH EDITION)

[625 .216]

Hydraulic buffers for railway vehicles. A mathematical analysis of their performance,

by L.B. BANKS M.I. Mech. E.

Increasingly under modern conditions the inadequacy of the standard spring buffer to absorb the shocks of shunting is being felt, particularly in mechanised marshalling yards where, if rapid working is to be achieved, occasional high speed impacts must be accepted. It can be shown that even under the most favourable conditions the maximum energy which can be absorbed by a spring buffer is only half that of a hydraulic buffer if the same maximum force is to be allowed. Much more important however is the fact that while the capacity of the spring buffer is soon exceeded even by quite modest speeds of collision the hydraulic buffer continues to absorb energy with the same efficiency until its bursting point is reached. Further, with a spring buffer the energy is merely stored to be released later but with the hydraulic buffer it is mainly dissipated in heat.

There are many different designs of hydraulic buffer but they all depend on one principle, that of forcing oil through an orifice or orifices whose size can vary with the stroke of the buffer. Sometimes this is achieved by means of a tapered stem passing through an orifice so that the annular space left will be decreased by the travel of the stem. An alternative design involves the successive blanking off of a number of ports, thus gradually reducing their effective area as the stroke increases. The force required to pass a liquid through a hole is proportional to the area and to the square of the velocity. It can be shown that in an efficiently designed buffer the area of the orifice should vary inversely as the remaining stroke of the buffer, in fact the following relationship should hold.

$$F = \frac{Mv^2}{2\,(L - s)} \quad \ldots\ldots \quad (1)$$

where F = force on a pair of buffers.
v = velocity
L = total stroke
s = travel
M = a design constant.

M is the " characteristic " of the buffer, it has the dimensions of mass and it can be shown that if a wagon is fitted with buffers so proportioned that M is equal to the mass of the wagon, then if that

wagon collides either with a rigid stop, or with any other wagon similarly proportioned, it will be brought to rest with constant deceleration, no matter what the initial velocity of the collision.

All practical hydraulic buffers are fitted with some form of return spring. These however absorb only a small proportion of the energy of collision except at the lowest speeds. Their only functions are to restore the buffers to their extended position and to keep the buffers extended when the train is on a down gradient. Their presence has been neglected in what follows :

Then with the aid of equation (1) the equations of motion can be written as :

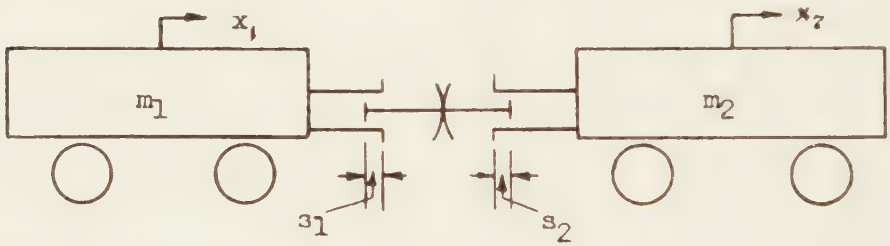
$$m_1 \cdot \ddot{x}_1 = \frac{-M_1}{2} L \frac{\dot{p}_1^2}{p_1} \quad \ldots\ldots\ldots\ldots \quad (2)$$

$$\frac{M_1}{2} L \frac{\dot{p}_1^2}{p_1} = \frac{M_2}{2} L \frac{\dot{p}_2^2}{p_2} \quad \ldots\ldots\ldots\ldots \quad (3)$$

$$m_2 \ddot{x}_2 = \frac{M_2}{2} L \frac{\dot{p}_2^2}{p_2} \quad \ldots\ldots\ldots\ldots \quad (4)$$

Substracting (2) from (4) gives :

$$\ddot{p}_1 + \ddot{p}_2 = \frac{M_1}{2\,m_1} \frac{\dot{p}_1^2}{p_1} + \frac{M_2}{2\,m_2} \cdot \frac{\dot{p}_2^2}{p_2} \quad \ldots \quad (5)$$



Let x_1 be the displacement of the 1st wagon;
x_2 be the displacement of the 2nd wagon;
s_1 be the travel of the buffers of the 1st wagon;
s_2 be the travel of the buffers of the 2nd wagon;
m_1 be the mass of the 1st wagon;
m_2 be the mass of the 2nd wagon;
M_1 be the characteristic of the two buffers of the 1st wagon;
M_2 be the characteristic of the two buffers of the 2nd wagon;
L be the total stroke of any buffer (assumed equal).

Also let $p_1 = 1 - s_1/L$ and $p_2 = 1 - s_2/L$.

and from (3): $\dfrac{\dot{p}_1^2}{\dot{p}_2^2} = \dfrac{M_2}{M_1} \cdot \dfrac{p_1}{p_2}$

$$\text{or} \quad \frac{\dot{p}_1}{\dot{p}_2} = \frac{dp_1}{dp_2} = \frac{M_2^{1/2}}{M_1^{1/2}} \cdot \frac{p_1^{1/2}}{p_2^{1/2}} \quad \ldots\ldots \quad (6)$$

Separating variables :

$$M_1^{1/2}\, p_1^{-1/2} \cdot dp_1 = M_2^{1/2}\, p_2^{-1/2} \cdot dp_2$$

Integrating :

$$M_1^{1/2}\, p_1^{1/2} = M_2^{1/2}\, p_2^{1/2} + C$$

at $t = 0 \quad p_1 = p_2 = 1$

$$\therefore C = M_1^{1/2} - M_2^{1/2}$$

which gives :

$$\frac{p_1^{1/2} - 1}{p_2^{1/2} - 1} = \frac{M_2^{1/2}}{M_1^{1/2}} \quad \ldots\ldots\ldots\ldots\ldots \quad (7)$$

Let $X_1 = p_1^{\frac{1}{2}} - 1 \quad X_2 = p_2^{\frac{1}{2}} - 1 \ldots$ (8) and (9)

then $\frac{X_1}{X_2} = \frac{\dot{X}_1}{\dot{X}_2} = \frac{\ddot{X}_1}{\ddot{X}_2} = \frac{M_2^{\frac{1}{2}}}{M_1^{\frac{1}{2}}} \ldots$ (10)

Differentiating (8) with respect to time t:

$$\dot{X}_1 = 1/2\, p_1^{-\frac{1}{2}} \cdot \dot{p}_1$$

hence $\dot{X}_1^2 = 1/4 \frac{\dot{p}_1^2}{p_1} \quad \ldots\ldots\ldots\ldots \quad (11)$

also
$$\begin{aligned}\ddot{X}_1 &= -\,1/4\, p_1^{-3/2} \cdot \dot{p}_1^2 \\ &\quad + 1/2\, p_1^{-\frac{1}{2}} \cdot \ddot{p}_1 \\ &= -\,p_1^{-\frac{1}{2}}\, \dot{X}_1^2 + 1/2\, p_1^{-\frac{1}{2}} \cdot \ddot{p}_1 \\ &= -\frac{\dot{X}_1^2}{1+X_1} + \frac{1}{2(1+X_1)} \cdot \ddot{p}_1\end{aligned}$$

or $\quad \ddot{p}_1 = 2\,\ddot{X}_1 (1 + X_1) + 2\,\dot{X}_1^2$

A similar relationship applies between p_2 and X_2.

Substituting the above in (5) gives :

$$2\,\ddot{X}_1 (1 + X_1) + 2\,\dot{X}_1^2 + 2\,\ddot{X}_2 (1 + X_2) + 2\,\dot{X}_2^2 = 2\,\frac{M_1}{m_1}\dot{X}_1^2 + 2\,\frac{M_2}{m_2}\dot{X}_2^2$$

Using (10) and collecting terms :

$$\ddot{X}_1 \left[X_1 \left(1 + \frac{M_1}{M_2}\right) + \left(1 + \frac{M_1^{\frac{1}{2}}}{M_2^{\frac{1}{2}}}\right)\right] + \dot{X}_1^2 \left[\left(1 + \frac{M_1}{M_2}\right) - M_1\left(\frac{1}{m_1} + \frac{1}{m_2}\right)\right] = 0$$

This is an equation of the form :

$$\ddot{X}_1 (aX_1 + b) + c\dot{X}_1^2 = 0$$

Substituting $\ddot{X}_1 = \dot{X}_1 \frac{d\dot{X}_1}{dX_1}$

we have $\frac{d\dot{X}_1}{dX_1} (aX_1 + b) + c\dot{X}_1 = 0,$

and separating variables :

$$\frac{d\dot{X}_1}{\dot{X}_1} = -\frac{c\,dX_1}{aX_1 + b}$$

Integrating :

$$\log_e \dot{X}_1 = -\frac{c}{a} \log_e (aX_1 + b) + \log_e A_1$$

where A_1 is an arbitrary constant.

$$\therefore \dot{X}_1 = A_1 (aX_1 + b)^{-c/a} \ldots\ldots\ldots (12)$$

$$\text{or} \quad \dot{p}_1 = 2\,A_1\, p_1^{\frac{1}{2}} \left[\left(1 + \frac{M_1}{M_2}\right)\left(p_1^{\frac{1}{2}} - 1\right) + \left(1 + \frac{M_1^{\frac{1}{2}}}{M_2^{\frac{1}{2}}}\right)\right]^{\dfrac{\frac{M_1}{m_1}\left(1 + \frac{m_1}{m_2}\right)}{1 + \frac{M_1}{M_2}} - 1} \ldots\ldots \quad (13)$$

by replacing M_1 with M_2, m_1 with m_2, A_1 with A_2 and p_1 with p_2 the expression for p_2 can be written down.

To evaluate A_1,

The initial conditions can conveniently be taken as :

$$\dot{X}_1 = u\,\dot{X}_2 = 0, \; t = 0 \text{ and } p_1 = p_2 = 1$$

(This can be made perfectly general by considering u to be the initial relative velocity of the two wagons.)

then $\quad \dot{p}_1 + \dot{p}_2 = \frac{-u}{L}$ at $t = 0$

and from (6) $\frac{\dot{p}_1}{\dot{p}_2} = \frac{M_2^{\frac{1}{2}}}{M_1^{\frac{1}{2}}}$

hence $\dot{p}_1 = - \dfrac{u/L}{1 + \dfrac{M_1^{1/2}}{M_2^{1/2}}}$ at $t = 0$

Substituting this in (13) gives :

$$A_1 = \frac{-u}{2L}\left[1 + \frac{M_1^{1/2}}{M_2^{1/2}}\right]^{-\frac{M_1}{m_1} \cdot \frac{1 + m_1/m_2}{1 + M_1/M_2}} \quad \dots\dots\dots\dots \quad (14)$$

and similarly for A_2.

The force between the two buffers is is given by (1) :

$$\text{i.e. } F = \frac{M_1 L}{2} \cdot \frac{\dot{p}_1^2}{p_1}$$

$$= \frac{M_1 L}{2} \cdot 4\,\dot{X}_1^2 \text{ (from 11)}$$

or by (12) :

$$= M_1 L \cdot 2\,A_1^2\,(aX_1 + b)^{-\frac{2c}{a}}$$

$$\text{Giving } F = \frac{M_1 u^2}{2L}\left[1 + \frac{M_1^{1/2}}{M_2^{1/2}}\right]^{-2\frac{M_1}{m_1} \cdot \frac{1 + m_1/m_2}{1 + M_1/M_2}}$$

$$\cdot \left[\left(1 + \frac{M_1}{M_2}\right)\left(p_1^{1/2} - 1\right) + \left(1 + \frac{M_1^{1/2}}{M_2^{1/2}}\right)\right]^{2\frac{M_1}{m_1} \cdot \frac{1 + m_1/m_2}{1 + M_1/M_2} - 2} \quad \dots\dots\dots \quad (15)$$

If wagon 1 has the " stiffest " buffers i.e. $M_1 > M_2$ then the buffers of wagon 2 will close first*, this follows from the relationship :

$$\frac{p_1^{1/2} - 1}{p_2^{1/2} - 1} = \frac{M_2^{1/2}}{M_1^{1/2}}$$

putting $p_2 = 0$, $p_1 = \left(1 - \dfrac{M_2^{1/2}}{M_1^{1/2}}\right)^2$

this gives the position of the 1st buffers when the 2nd buffers are closed ($p_2 = 0$). After this point equation (15) no longer applies. It should be noted that the position depends on M_1 and M_2 and not on the masses of the wagons or their relative velocities of collision.

The force acting after one pair of buffers has closed can be found by letting M_2 in equation (15) $\rightarrow \infty$ which gives :

$$F_1 = \frac{M_1 u_1^2}{2L}\, p_1^{\frac{M_1}{m_1}\left(1 + \frac{m_1}{m_2}\right) - 1} \quad \dots\dots \quad (16)$$

where u_1 must be chosen so that $F = F_1$

when $p_1 = \left(1 - \dfrac{M_2^{1/2}}{M_1^{1/2}}\right)^2$

i.e. when the 2nd buffers are on the point of closing, which is given by putting $p_1 = (1 - M_2^{1/2}/M_1^{1/2})^2$ in equation (15) :

$$\text{i.e. } F = \frac{M_1 u^2}{2L}\left(1 + \frac{M_1^{1/2}}{M_2^{1/2}}\right)^{-2\frac{M_1}{m_1} \cdot \frac{1 + m_1/m_2}{1 + M_1/M_2}} \left(1 - \frac{M_2^{1/2}}{M_1^{1/2}}\right)^{2 \cdot \frac{M_1}{m_1} \cdot \frac{1 + m_1/m_2}{1 + M_1/M_2} - 2} \quad \dots\dots\dots\dots\dots \quad (17)$$

* A case might be made out for proportioning the total travel L of the buffer to the characteristic M, thus ensuring simultaneous closing of all buffers.

and immediately after closing by putting the same value for p_1 in (16) :

$$F_1 \quad \frac{M_1 u_1^2}{2L}\left(1 - \frac{M_2^{1/2}}{M_1^{1/2}}\right)^{2\frac{M_1}{m_1}\left(1+\frac{m_1}{m_2}\right)-2} \qquad \ldots\ldots\ldots \quad (18)$$

at the point of closing $F = F_1$ therefore by equating (17) and (18) :

$$u_1^2 - u^2\left(1 - \frac{M_1^{1/2}}{M_2^{1/2}}\right)^{-2\frac{M_1}{m_1}\cdot\frac{1+m_1/m_2}{1+M_1/M_2}} \cdot \left(1 - \frac{M_2^{1/2}}{M_1^{1/2}}\right)^{2\frac{M_1}{m_1}\left(1+m_1/m_2\right)\left(\frac{1}{1+M_1/M_2}-1\right)}$$

and substituting this in equation (16) gives an expression for the force between the buffers after one buffer has closed, i.e. :

$$F_1 = \frac{M_1 u^2}{2L}\left(1 + \frac{M_1^{1/2}}{M_2^{1/2}}\right)^{-2\frac{M_1}{m_1}\cdot\frac{1+m_1/m_2}{1+M_1/M_2}} \cdot \left(1 \quad \frac{M_2^{1/2}}{M_1^{1/2}}\right)^{2\frac{M_1}{m_1}\left(1+\frac{m_1}{m_2}\right)\left(\frac{1}{1+\frac{M_1}{M_2}}-1\right)}$$

$$\cdot\, p_1^{\,M_1/m_1\,(1+m_1/m_2)-1} \qquad \ldots\ldots\ldots\ldots\ldots \quad (19)$$

Reverting to equation (15) this may be used to find the force when a wagon collides with a rigid stop by letting both M_2 and $m_2 \to \infty$, giving :

$$F_r = \frac{M_1 u^2}{2L} p_1^{\frac{M_1}{m_1}-1} \qquad \ldots\ldots\ldots \quad (20)$$

it will be seen that F_r is constant and independent of the buffer travel if :

$$M_1/m_1 - 1 = 0$$
$$\text{i.e. } M_1 = m_1$$

that is when the " characteristic " of the buffers is matched to the mass of the wagon.

The effect of collision between two similar wagons can be found by putting $M_1 = M_2 = M$ and $m_1 = m_2 = m$ in equation (15) then :

$$F_e = \frac{Mu^2}{8L} \cdot p_1^{M/m-1} \qquad \ldots\ldots\ldots \quad (21)$$

and again the force is constant and independent of buffer travel when $M = m$, the actual force however being 1/4 what it is when the collision is with a rigid stop.

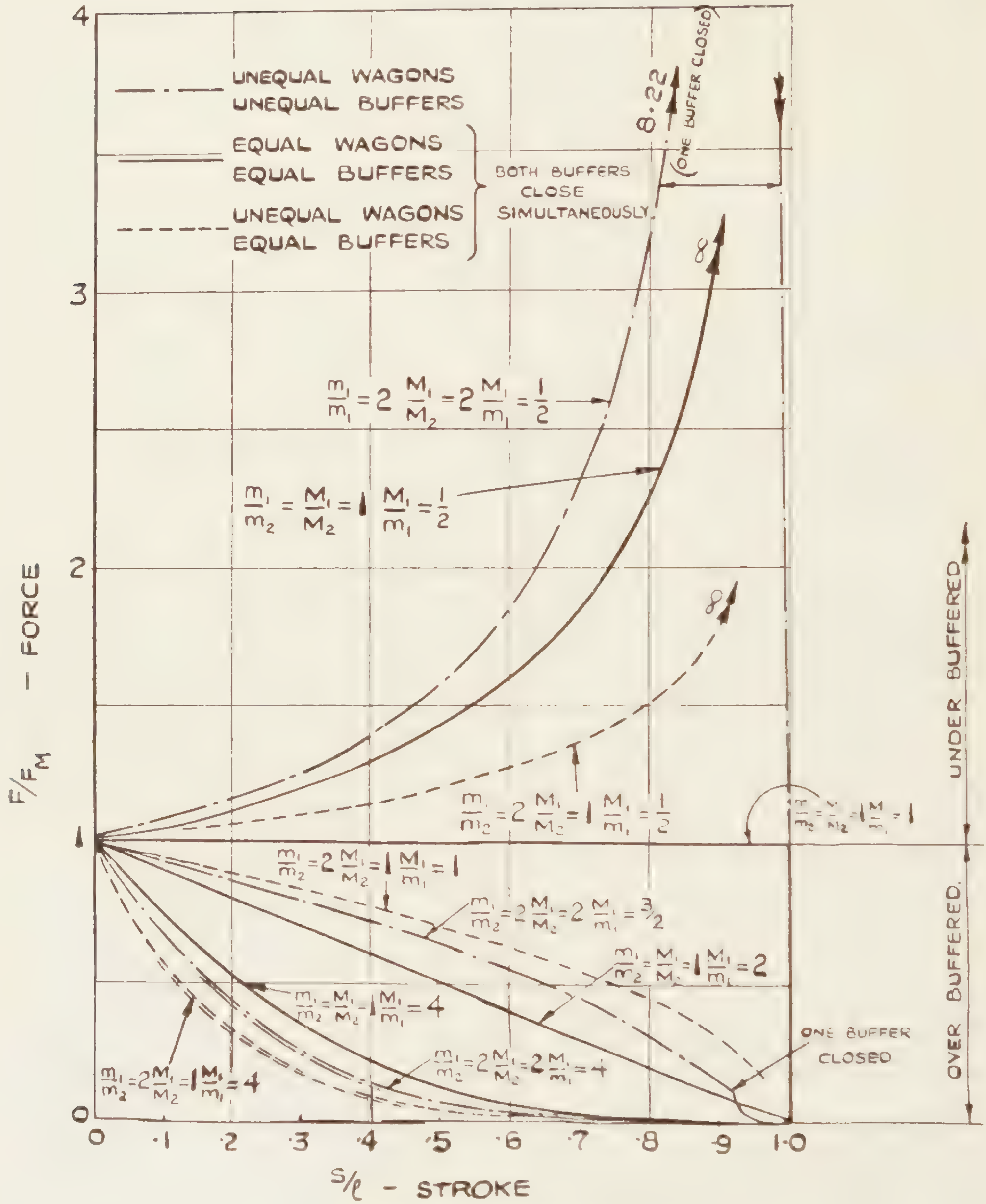
In the general case of unequal wagons it will be seen that the buffer force is constant when the exponent :

$$2\frac{M_1}{m_1}\frac{1+m_1/m_2}{1+M_1/M_2} - 2 = 0$$

$$\text{i.e. } \frac{1}{M_1} + \frac{1}{M_2} = \frac{1}{m_1} + \frac{1}{m_2}$$

there are of course an infinite combination of values which satisfy this equation but it is satisfied if $M_1 = m_1$ and $M_2 = m_2$*, but these have already been found to be the required characteristics for the buffers

* This relationship was discovered by D.L. Turner M.A. a colleague of the author, he called buffers so matched " tailor made " buffers.



STROKE/FORCE RELATIONSHIP
FOR DIFFERENT BUFFER CHARACTERISTICS AND WAGON MASSES

of each individual wagon for collision with maximum efficiency into either similar wagons or into a rigid stop. If all wagons had constant mass then the ideal would be for them each to be fitted with buffers to match. Unfortunately this solution is not practical, first because wagons are not of constant mass but run both full and empty and secondly, there is such a wide range of wagons that it would not be practicable to fit every type with buffers to suit.

The force exerted when the wagons are fitted with matched buffers is given by :

$$F_m = \frac{M_1 u^2}{2 L}\left(1 + \frac{M_1^{1/2}}{M_2^{1/2}}\right)^{-2} \quad \text{from equation (15)}$$

and

$$\frac{F}{F_m} = \left[\frac{1 + M_1/M_2}{1 + M_1^{1/2}/M_2^{1/2}}\left(p_1^{1/2} - 1\right) + 1\right]^{2\frac{M_1}{m_1} \cdot \frac{1 + m_1/m_2}{1 + M_1/M_2} - 2}$$

The performance of the buffers therefore requires consideration of the three parameters :

$$M_1/M_2,\ m_1/m_2 \text{ and } M_1/m_1$$

all combinations of wagons and buffers having the same values of these three parameters will give the same values for F/F_m, but not of course the same absolute values of force, displacement velocity, etc. The figure shows the effect of varying these three parameters. It will be seen that the masses of the wagons enter into either the equation for the velocity (13) or the equation for the force (15) only in the exponent and if the masses m_1 and m_2 are replaced by m_3 and m_4 then if :

$$\frac{M_1}{m_1}\frac{(1 + m_1/m_2)}{1 + M_1/M_2} = \frac{M_1}{m_3}\frac{(1 + m_3/m_4)}{1 + M_1/M_2}$$

$$\text{i.e. } \frac{1}{m_1} + \frac{1}{m_2} = \frac{1}{m_3} + \frac{1}{m_4}$$

then the forces and the velocities of buffer closure arising from the collision are identical. This relationship is very useful when buffers are being tested.

[**656** .211 .5 (492)]

The platform roofs of the new Rotterdam Central Station,

by S. Noyon,

Engineer of the S. A. Spoorwegopbouw (Holland).

After the last war, the Netherlands Railways carried out a project that had been laid aside for a long time, the linking up of the Rotterdam Maas station, the terminus of the lines from the East, with that of Rotterdam Delftsepoort on the north to south artery.

The most practical solution seemed to be to build the new central station on the site of the Delftsepoort station.

Fig. 1.

The six platforms which, owing to the road services, had to be about 3 m above the level of the existing platforms occupied the whole of the site, so that

their construction, laying the tracks and putting them into service had to be carried out in successive stages. In addition, the work could only be commenced after the construction of new installations for the freight traffic and shunting services.

Fig. 2.

As a result between 1953 and 1954, the platform on the north side was put into service, then in 1955 and 1956 the second platform; it is expected that the new raised up third platform will be completed during 1957. These three platforms can deal with the whole of the traffic, althought conditions will still be very restricted

The new main building as well as the adjoining platform will also be put into service during 1957, after which it will be possible to start work on the remaining two platforms.

Owing to the fact that the platforms had to be constructed one at a time, an individual roof had to be built for each platform instead of the usual structure covering all the lines and platforms. The short fixed intervals between successive train départures also made it less necessary to provide complete protection for passengers

The platforms which are 12 m wide and 300 m long are covered in over a length of 200 m. In order not to impede circulation along the trains, the only supports have been arranged along the centre of the platforms. As the roofs

must be very high over the track in order to allow sufficient light to come in over the tops of the trains, and on the other hand, they need not be particularly high in the centre of the platform, the design which appeared the most suitable was that of a curved V and this was adopted.

Fig. 3.

In addition, as it was necessary to space the supports very far apart, a shell-like concrete structure was adopted (see fig. 1).

These shells, made up of arched plates 7 m in radius, form beams of one or two bays of 13.50 m length with 5.50 m cantilever extensions at the two ends so that the positive and negative stresses are about the same. Expansion joints have been provided at the ends of the beams. A reinforcing wall is fitted on the upper side of the roof, above each support. The shell, which has twofold reinforcement, is 8 cm thick.

Artificial lighting is by means of fluorescent tubes with reflectors arranged in line along the lower edge of the glazed front of the windscreens which gives excellent indirect lighting by reflection from the white roofs (fig. 2).

All the electric cabling has been laid inside the reflectors and in wooden covering along the joints of the roof shells (see fig. 3). The space between some of the supports have been closed by partitions in which seats are provided. Above the underground entrance gallery, there is a waiting room.

In order to avoid unequal heating through reflection of the sun's rays, the northern wing has been insulated with cork on the upper portion.

It would no doubt be possible to use the principle of V shaped shells for making much wider roofs than these, especially if prestressed concrete is used. However, in the present case, it was not considered necessary to increase the distance between supports.

The two wings covering the first platform, which adjoins the passenger buildings, have been erected one behind the other in such a way as to form a shed.

The calculations for shells of V form is more or less the same as that required for sheds. It is not necessary to have a heavy edge to assure the solidity of the structure. All the same this does contribute to the rigidity of the whole and plays an important part in the attachment of the glazed screens, as well as the aesthetic appearance of the structure.

[625 .214]

Roller bearing axleboxes,

by Dipl.-Ing. Joseph FIJALKOWSKI,

of the Polish State Railways.

The roller bearing axleboxes used up to the present on railway rolling stock in principle have two sets of rollers fitted side by side on the journal or more recently a single row with spherical rollers. The body of the box in this case lies directly under the springs which transmit the load to it.

Besides these two systems, boxes with special rollers have been developed over a number of years for use in rolling stock.

The particular feature of these boxes is the use of one outer and one inner race, each in one piece, between which there are two rows of cylindrical rollers.

For more arduous working conditions and higher speed, this type of bearing is fitted with two additional rings to withstand the axial shocks and horizontal transverse stresses.

The " Unité Technique " under the auspices of the I.R.U. (U.I.C.) has approved the use of roller bearing boxes with two rows of rollers in single piece races using cylindrical, conical or spherical rollers.

Axleboxes with single piece races are used on a large scale by the American Railways : these boxes are of the Hyatt and SKF types illustrated on the 19th edition of the *Car Builders Cyclopedia* of 1953, pp. 1060, 1061, 1062 and 1063. Similar boxes of Hoffmann make are used on a large scale by British Railways. These boxes are described in an excellent edition of the catalogue " Hoffmann Parallel Roller Bearing Axleboxes ".

Axleboxes with single piece races mentioned above are used widely on the PKP (Polish State Railways) on carriages, goods wagons, electric locomotives, steam locomotives and tenders. The axleboxes in question have the trade mark PKP and are shown in figures 1, 2, 3, 4, 5 and 6. These designs show clearly that provision has been made to fit to the carriages, locomotives and tenders, roller bearing axleboxes of type PKP 26, PKP 30, PKP 34 and PKP 48 fitted with axial supporting bearings with cylindrical rollers.

For easier service conditions, that is to say, on standard gauge goods wagons as well as on all goods wagons and carriages for the narrow gauge lines and tramways type PKP 16 u, PKP 24 u and PKP 26 u with one piece races and no supporting bearing have been manufactured.

The type PKP roller bearings have been designed after the Hyatt and Hoffmann whereas the radial bearings have been based on the bearings specified in the McGill catalogue " Precision Bearings " of 1940, p. 26.

The PKP bearings have been in use on the Polish State Railways to the number of 5 000 for the last ten years.

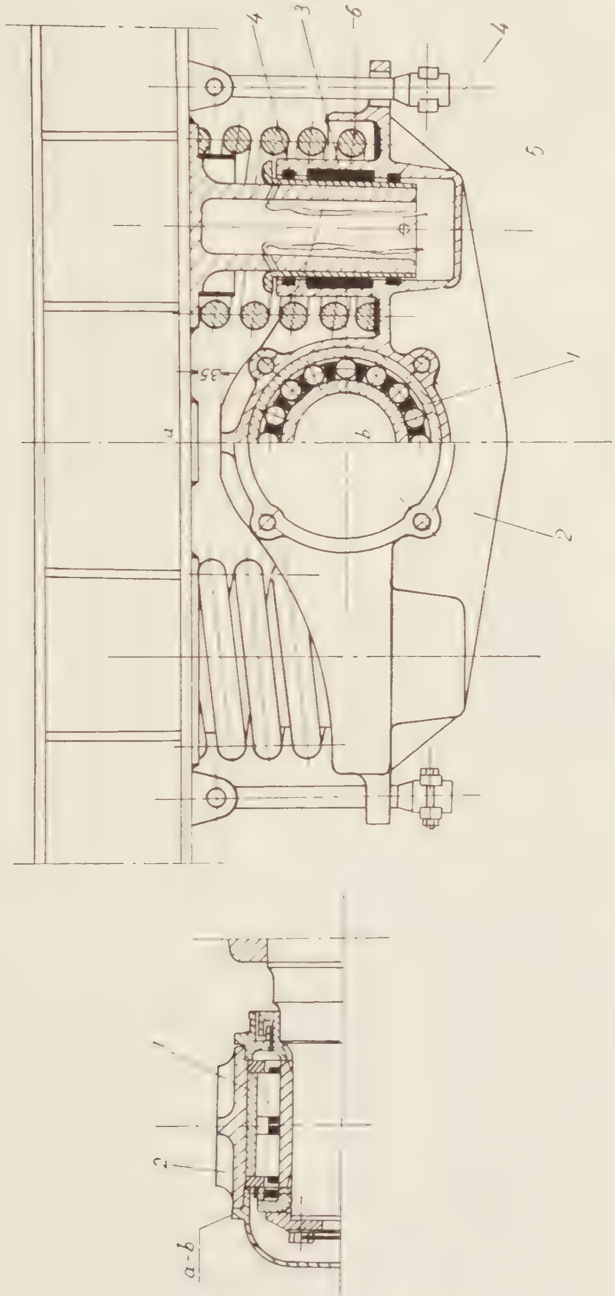


Fig. 1. — **PKP** roller bearing axlebox guided between two cylindrical guides.

1. Race.
2. Body of box.
3. India rubber.
4. Packing.
5. Stop.
6. Spring.

The graph of figure 10 shows that the PKP bearings have a very much longer life than the other classical type of bearings (two races in one box).

Owing to the simplicity of construction of the different parts and ease of application of single piece races the PKP bearings have given no trouble in manu-

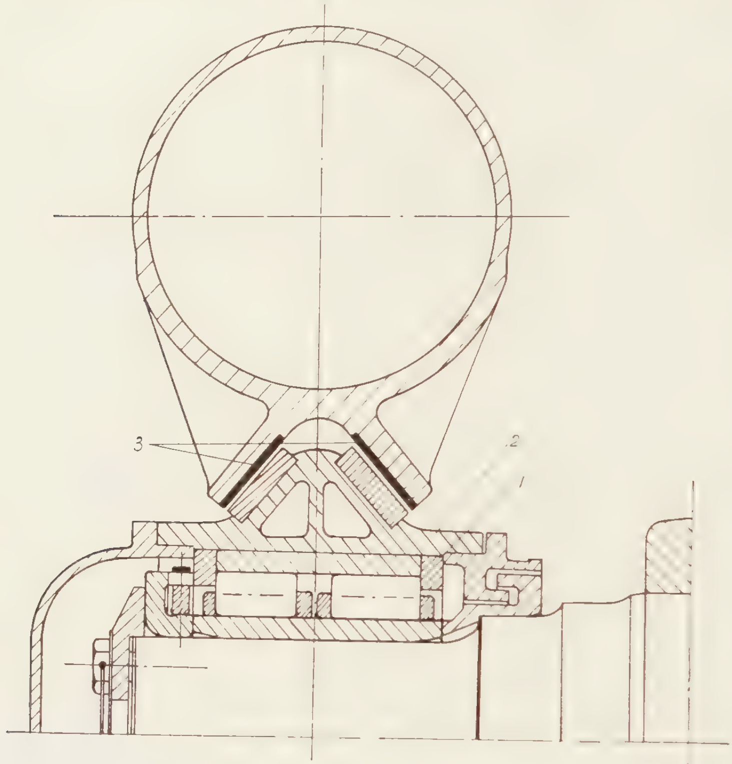


Fig. 2. — Roller bearing axlebox guided between two slides.
1. Race. 2. Body of box. 3. Slides.

The following reasons for the greater wear of the classic types of bearings have been observed :

flaking of the roller path — 80 %;

formation of crevices on the faces of the races — 10 %;

damage to the cages (wear of cages of spherical rollers and broken rivets in the cages of the bearings with cylindrical rollers) — 8 %;

other causes — 2 %.

facture. Actually, the races and bearings rings are made from drawn tubes of Cr Ni steel cut off in suitable machines. The cages are case-hardened castings in aluminium bronze. The bores are finished machined after casting in a boring machine. The support bearing cages are made of aluminium bronze, machined and then riveted to the outer steel ring. The surfaces of the PKP races are finely polished (superfinish) and undergo a

precision grinding. Each bearing is tested under maximum load and speed for one hour.

The superiority of the bearings with one piece rings reinforced if need be by a support bearing is due in our opinion to the following facts (see fig. 11) :

1. The use of a support bearing increases the life of the bearing as expressed as an increase in distance travelled by the formula :

$$L = \left(\frac{C}{P}\right)^{10/3}$$

because the equivalent load P is equal in this case solely to the radial load, without a lateral component : the supporting bearings with cylindrical rollers show no visual wear after running 1.5 million kilometres;

2. The application of one piece rings simplifies the manufacture of the race and reduces the pressure on the edges of the rollers. The high loads observed at the points marked "×" in figure 11 result from the tolerance of play allowed when manufacturing the rings and the rollers.

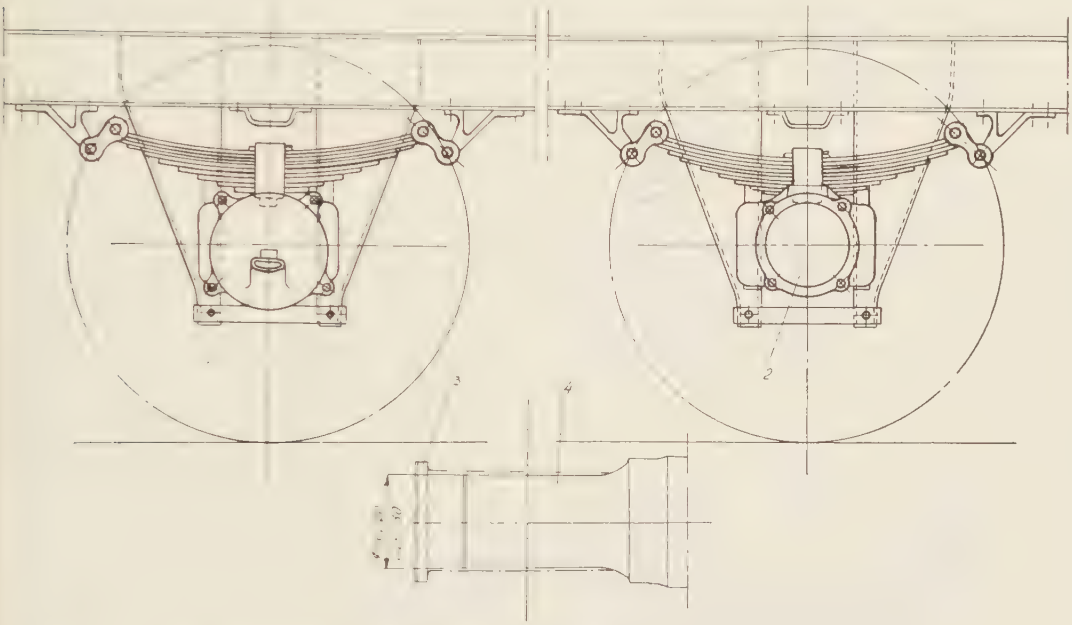


Fig. 3. — The replacement of plain bearings by roller bearings on goods wagons and four and six-wheeled carriages.

1. Plain bearing.
2. Roller bearing.
3. Journal for plain bearing.
4. Journal for roller bearing after turning and polishing the journal shown in 3.

As an example in the case of cylindrical roller bearings with dimensions of 130 × 240 × 80 mm (fig. 11) the following tolerances are allowed :

— on the size of the rollers after selection . . . . . . 0.002 mm;

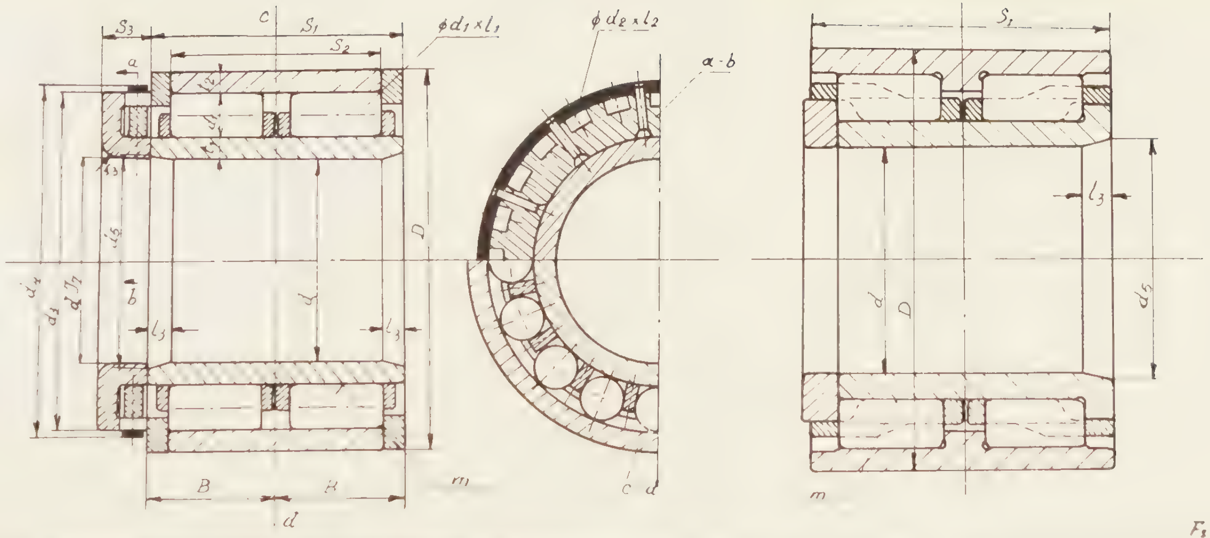


Fig. 4

Seria	d	B	S_1	S_2	S_3	l	t_1	t_2	r_1/r_2	l_3	D	d_1	d_2	l_1	l_2	n_1/n_2	d_3	d_4	d_5	C_1 kg	C_2 kg	m luz
PKP16u	80	58	116	96	—	15	12	11	1,5/1	—	170	22	22	42	42	30/	—	—	84	50 000	—	od 0,160 do 0,260
PKP24u	120	73	146	119	—	15	14	12	1,5/1	—	220	24	24	50	50	32/	—	—	124	80 000	—	,,
PKP26u	130	80	160	133	—	15	14	13	1,5/1	—	240	28	28	58	58	32/	—	—	134	100 000	—	,,
PKP26	130	90	160	133	30	15	14	13	1,5/1	15	240	28	16	58	11	32/24	214	224	134	100 000	11 300	,,
PKP30	150	88	176	142	37	15	17	15	1,5/1	15	280	34	20	60	12	32/24	252	264	155	130 000	16 000	od 0,170 do 0,260
PKP34	170	120	240	200	49	25	25	20	2,5/1	20	360	50	28	86	16	26/22	317	333	186	230 000	17 300	od 0,180 do 0,270
PKP48	240	74	148	118	53	15	25	19	1,5/1	20	400	36	28	50	16	44/28	372	388	216	158 000	25 800	od 0,200 do 0,300

Fig. 4. — PKP roller bearing axleboxes. Principal dimensions proposed for general use.

N. B. — Lug = play. — Od 0.160 do 0.260 = from 0.160 to 0.260.

(Corrigenda to the above table)

Column d_5 : The last number 216 is to be replaced by *246*. — The *three first numbers* 96, 119, 133 of Column S_2, 22, 24, 28 of Column d_2 and 42, 50, 58 of Column l_2 *must be deleted*.

— on the diameter of the inner race . . . . 0.025 mm;
— on the outer race . . . . 0.035 mm;
— on the outside diameter of the outer race. . . . . . 0.040 mm;
— on the outside diameter of the inner race. . . . . . 0.040 mm.

$$tg\ \alpha 3 = \frac{0.035}{80};\quad tg\ \alpha 4 = \frac{0.040}{80}$$

and $$tg\ \alpha 5 = \frac{0.040}{80}.$$

If we suppose the roller under load should bear over its whole length on

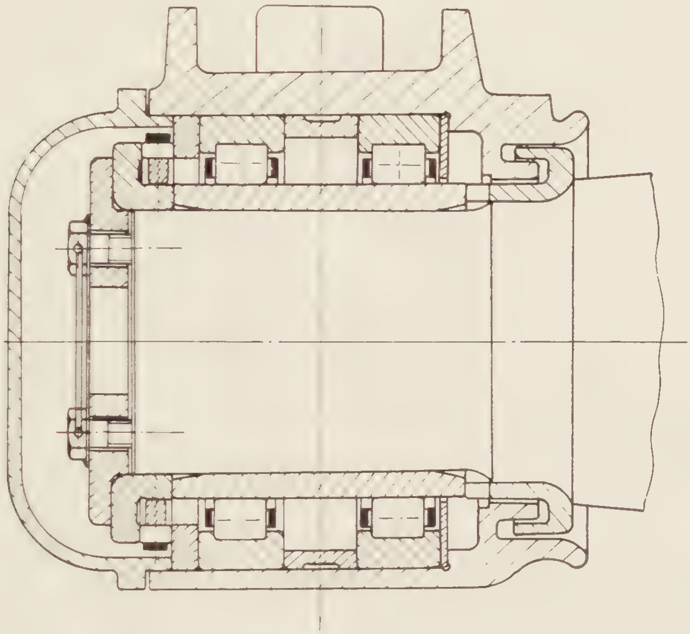


Fig. 5. — Hoffmann axlebox with supporting bearing using cylindrical rollers equivalent to the PKP boxes.

In the case when the tolerances are at their maximum values as represented in figure 11 the angle α causing the rollers to assume the oblique position would have approximately the value :

$$\alpha = \alpha 1 + \alpha 2 + \alpha 3 + \alpha 4 + \alpha 5$$

or for

$$1 = 80\ \text{mm}\ tg\ \alpha 1 = \frac{0.002}{80};\quad tg\ \alpha 2 = \frac{0.025}{80};$$

the face of the race, the maximum deformation of the path and of the roller of length $b = 80$ mm in the bearing of figure 11 can be determined by the following formula :

$$S_1 = \frac{50}{80}(0.002 + 0.025 + 0.035 + 0.040 + 0.040) = 0.0665.$$

For the PKP bearings the above values would be :

$$tg\ \alpha = \frac{0.002}{80};$$

$$S_2 = \frac{50}{80} \times 0.002 = 0.00125 \text{ mm.}$$

Consequently, the deformations at the edges of the rollers of the bearing shown in figure 11 would be $\dfrac{0.0665}{0.00125} = 52.3$ times more than those of a PKP bearing.

The deformations in question result in an increase in pressure on the edges of the rollers which cause an increase in the loads at the edges according to the well known formula :

$$\sigma = 995 \cdot \sqrt{\frac{P}{Cp \cdot dp}} \text{ kg/cm}^2.$$

Consequently, the life of a roller bearing of this type would be shortened appreciably.

Fig. 6b.

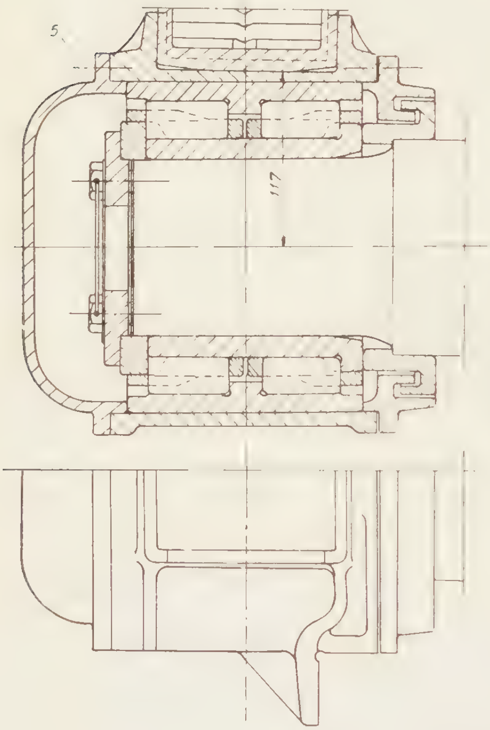


Fig. 6a.

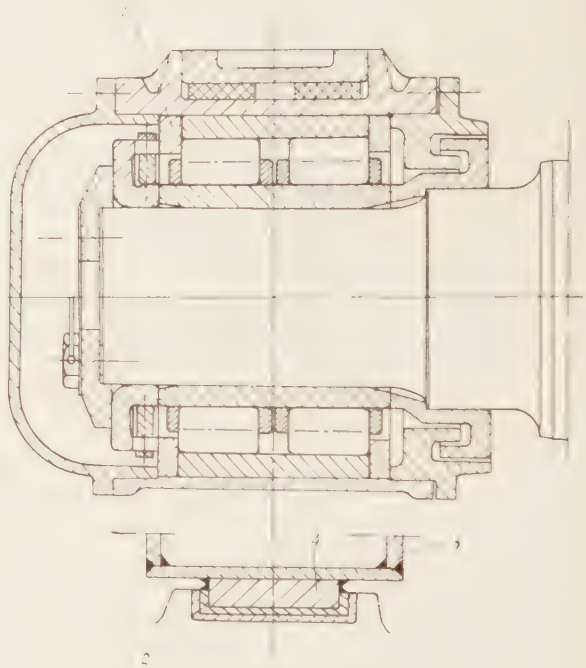


Fig. 6. — PKP roller bearing in a single box.

1. PKP roller bearing with supporting roller bearing for heavy duty (fig. 6a).
2. Guides of the axlebox on both sides.
3. Bogie frame.
4. Slides or guides.
5. PKP roller bearing with supporting bearing of plain type for lighter duty (fig. 6b).

As regards the axlebox bearings of the PKP type of figure 3, the distance between the axlebox slides should be selected in relation to the distance between the two boxes on an axle with a tolerance lying between 2.4 mm and 1.5 mm. The runs steadily with axial plays of 5 mm at the sealing labyrinth. At the same time, the transmission of the supplementary load due to unavoidable errors in fitting up the bogie is eliminated.

The labyrinth packing on the wheel

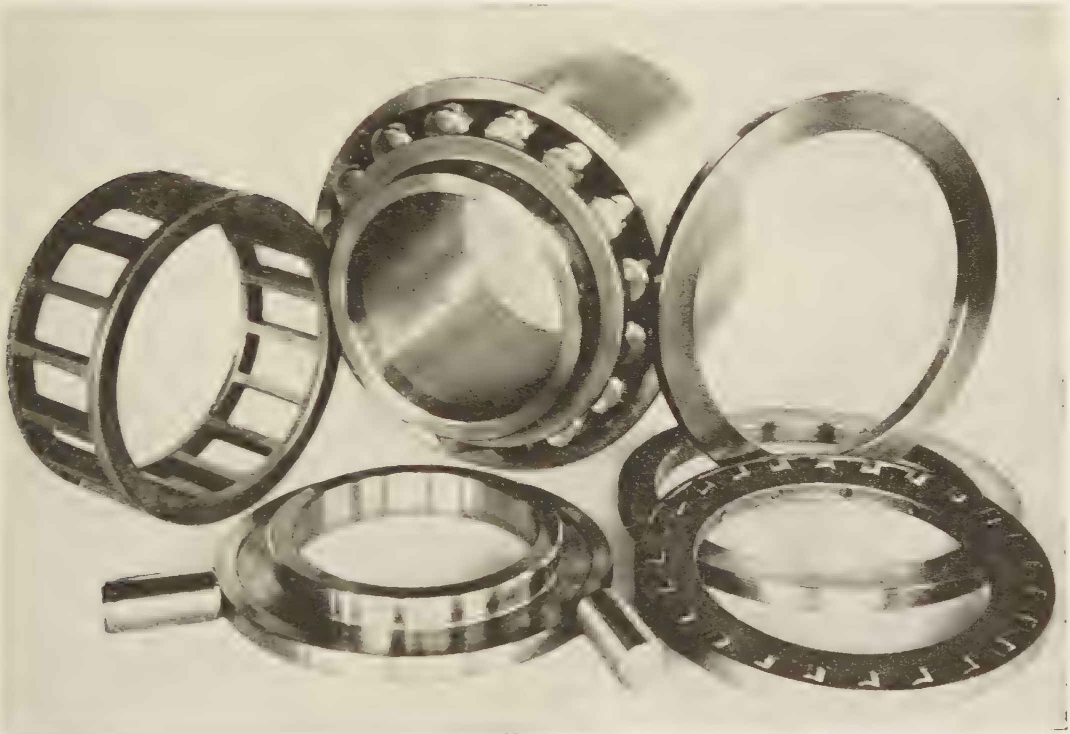


Fig. 7. — Dismantled PKP 26 roller bearing axlebox.

longitudinal play in each support bearing will be 0.75 to 1.2 mm. It is recommended practice to guide the roller bearing box on the two sides or by levers (see fig. 6). At the same time, it is good practice to place a packing in hard rubber or in wood between the spring buckle and the body of the axlebox. When this method is used in fitting PKP bearings, the bogie boss side plays here the role of the support bearing of plain bearings.

We recommend the roller bearing of figures 3 and 6*b* for four-wheeled goods wagons. The body of the box is well supported by the buckle of the spring thereby making it possible to maintain the dimension of 117 mm (see fig. 3 and 6*b*) as with the plain bearings. At

the same time, it becomes possible to replace the existing plain bearings on the goods wagons without rebuilding these latter. All that is needed is to remachine and finish the journals of the existing axles by bringing them to the diameter of 100 mm or 120 mm.

In our opinion the PKP axleboxes or those of similar design should be accepted by the I.R.U. (U.I.C.) as these are relatively easy to manufacture even in countries with little industrial development. Their production requires a small quantity of material of inferior quality. The life of these bearings is remarkable.

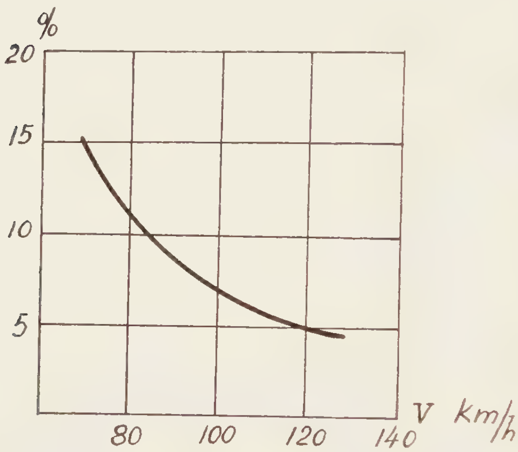


Fig. 8. — Graph showing the reduction in tractive effort obtained by using roller bearings (saving of tractive effort as a function of the speed in km/h).

The acceptance of a standard bearing by the railways on an international scale would permit, we feel, a quicker rate of equipment of rolling stock with roller bearing axleboxes. From the point of view of the manufacture of axleboxes the acceptance of the standard axlebox would cause only the starting up in the different countries of specialised work on automation lines to the fullest extent and equipped with standardised machine tools using standard components as well as standard installations essential for high production.

The advantages resulting from the application of roller bearings to railway stock can be summed up as follows :

1. Reduction of the coefficient of resistance both at starting and when running of the vehicles and consequently improvement in the use of the tractive effort available and a reduction in fuel consumption.

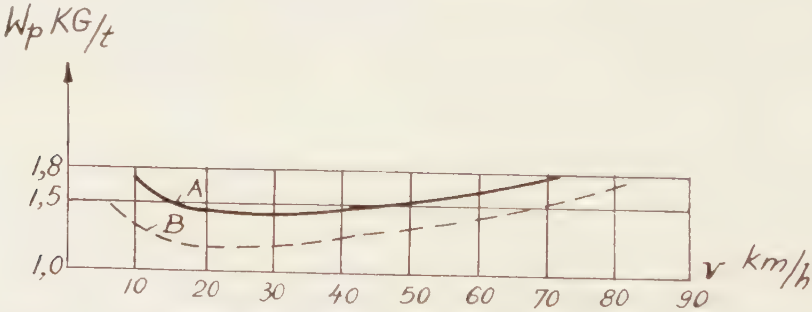


Fig. 9. — Reduction in rolling resistance due to the application of roller bearings.

(A) Resistance with plain bearing.

(B) Resistance with roller bearings.

The vehicles were run 300 km at a specified speed before making the trial runs.

2. Reduced consumption of lubricants.

3. Saving of alloys used on plain bearings.

4. Lower costs of repairs.

5. Reduction in the number of hotboxes resulting in railway vehicles being withdrawn from service.

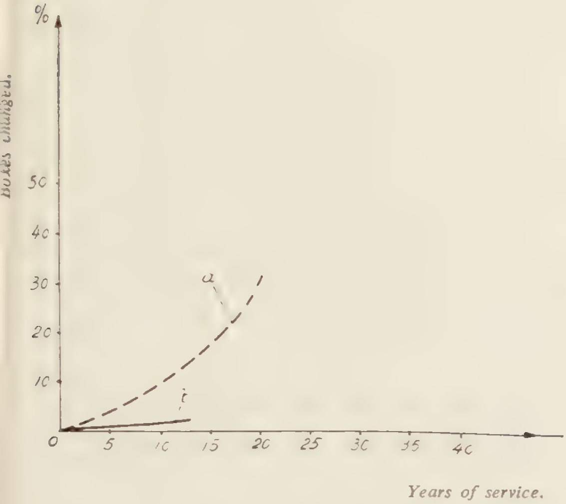


Fig. 10. — Wear of roller bearings as percent.

(*a*) Standard (catalogue) bearings with separate races (two bearings in one box).

(*b*) Single race roller bearings with supporting bearing (PKP type).

The influence of roller bearings on the reduction of the coefficient of rolling resistance on railway vehicles is calculated in different ways in technical literature.

For example, the graph 2 published in the *Bulletin of the International Railway Congress Association*, No. 9, 1956, underlines the saving in tractive effort according to data collected in Western Europe.

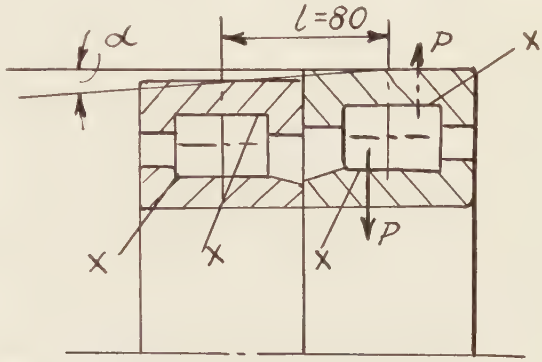


Fig. 11. — Axlebox with roller bearings of a normal type (catalogue).

The graph 9 for the Sovietic publication " Steam Locomotives ", by G.R. Syromiatnikow, Moscow, 1949, p. 680, gives the reduction in rolling resistance of roller bearing fitted vehicles according to results obtained by the Institute of Tests.

To sum up, it can be expected in accordance with the reflections given above, that the roller bearings will be a source of savings of fuel, of constructional materials, of lubricants, and of labour for all the railway systems of the world.

[625 .251]

Recent developments in driver's brake valves,

by A. NIEVERGELT.

Engineer, Swiss Federal Railways, Berne.

1. Historical summary.

The appearance of new types of brake distributors necessarily should mean a parallel development in the driver's valves to take full profit of the advantage of the new brakes. Let us recollect that on examples the old Westinghouse No. 4 driver's valves and Knorr valves which most locomotives still have were made at a time when the moderable or graduable release of the brake was scarcely known. It is no more than some thirty years since the new triple valves enabled the release to be graduated and necessitating the driver's valves being modified to provide additional features, in particular :

(*a*) Greater capacity to give faster release;

(*b*) Making good losses also when in the brake applied position;

(*c*) Finer graduation.

Actually, the distributors with graduable release only release when the former pressure is re-established in the train pipe and in the reservoirs. This required per unit of time a higher rate of feed than in the case of the brake not graduable during release. The driver's valve therefore must have greater capacity to meet this condition. Up to a few years ago, there was general criticism of the long time to release the brakes on trains with brakes having graduable release. Certain railways even felt the need to lower the release pressure at the expense of the field of regulation. By making new distributors protected against excess pressure the condition needed to permit higher feeds of air were provided : the release in the case of long trains can be shortened considerably.

At the present time, brake technique frequently allows the brakes in a long rake fitted with modern distributors, to be released more quickly than a similar rake with non-moderable release distributors and whose reservoirs are uncompletely refilled after release as the two processus — release and refill — are not necessarily linked together as is the case with the brakes graduable during release. The long rakes with modern distributors giving moderable release release themselves in a lapse of time but little different from that of a single vehicle.

Moreover, to be able to maintain a constant braking effort and prevent the exhaustion of the brake power on long down gradients of constant declivity the almost unavoidable leaks must be made good. For this to be done the driver's brake valve too must be of a type to make good these leakages with the handle in the brake on position.

These questions had already been raised about 1930, that is to say, on the introduction of brakes with graduable release (Drolshammer, Bozič, Hik and Breda). To begin with a number of improved driver's valves of greater capacity were introduced. These retained in principle the system of the old valves with first of all an increase in the compensation of leakage in the running position. A number were also provided at that time with a neutral position with leakage compensation during braking.

A fundamentally new driver's brake valve was introduced by Bozič followed by the type C Knorr. The Bozič was characterised by the use, we believe for the first time, of a pressure regulator controlled by the handle of the driver's valve. To each position of this latter, there was a predetermined pressure in the train pipe. The type C Knorr valve went still further and introduced the automatic limitation of the duration of the brake filling surge (l'à-coup de remplissage). The two valves however were frequently out of order and made problematical the replacement of the older simpler valves with which the staff was familiar.

2. Evolution during the last ten years.

At the end of the second world war, a new phase took place in the development of the brakes due to the application of new constructional principles. The slide valves were replaced by valves and the piston rings by diaphragms in india rubber. Great progress too was made in mastering the dynamic pneumatic processes in a better way than before. The brakes became more sensitive and consequently the speed of propagation was increased considerably. At the present time a depression of 0.05 kg/cm^2 in the train pipe is enough to actuate the brake on the last vehicle of a rake of 150 pairs of wheels whereas a depression of 0.3 kg/cm^2 was required with the old triples i.e. those with slide valves and piston with leather sleeves. The great sensitivity enabled too a finer regulation of the braking effort. In order however to ensure that the triples through their very sensitivity did not react uncontrolably through minimum fluctuation of pressure in the train pipe, it became necessary to have a driver's valve which would allow very fine adjustments of pressure to be made. This lead to a new driver's valve being perfected using details proved in service in the new triple valves and in particular valves with hard rubber and cam controlled. The high sensitivity required during the regulation of the brake power progressive variations of pressure which was most successfully achieved with indirect pneumatic operation. The same exigencies lead in addition to a method of automatic or semi-automatic operation as slow and progressive variation are difficult to get by hand, for example by the slow movement of a handle valve.

Tightness of a valve is more easily achieved by contact « metal on rubber » than by contact « metal on metal »; it involves less upkeep seeing that no lubricator is needed nor is there any grinding of seats or cones. The same remark applies to diaphragms and slide valves may be expected to disappear.

3. New features of the modern driver's brake valve.

(a) Operation of the pressure regulator.

A particular feature to be pointed out is the dependence between the position of the valve handle and the pressure in the train pipe found with the Bozič, Knorr-Automatic C and D and the Oerlikon FV 3 and FV 4. The Germans call this type of driver's valve « Selbstregler », that is to say, self regulating seeing that the pressure in the train pipe is controlled solely by the position of the handle. The French express this feature by the name « robinet manométrique » (pressure gauge valve) because of the direct dependence between the position of the handle and the indication on the pressure gauge.

Two different types of driver's valves are thus indicated — one regulates the pressure in the train pipe — on application as on release — by the to and fro movement of the handle, that is to say be opening and closing during a predetermined interval of time a « valve » which allows a certain volume of air to escape or enter. The other regulates the pressure by a more or less large movement of the handle which acts on the spring of the pressure regulator. The position of the handle automatically determines the required pressure in the train pipe. The manipulation of this type of driver's valve is simpler and makes the driver's duty easier. With the indirect control of the pressure in the train pipe, the influence of the length of the train and of other factors on the method of using the driver's valve handle is eliminated.

(b) Brake filling surges.

Where, when, and how to give a brake filling surge is a question much discussed. Everyone seeks how to release the brakes quickly, and especially brakes which can be graduated during release. By enlarging the release orifice in the driver's valve, a large quantity of air can pass but this method is only efficacious when there is a difference of pressure. Ultimately, it must be admitted the only way to send sufficient air to the rear of the train is by increased pressure. For this air to be in sufficient quantity in the last stages of the release the working pressure at the head of the train has to be exceeded momentarily. The brake filling surge reduces by about one half the time to release the brakes on a long train, comparatively to the time taken with the driver's valve in the running position. Those initiated in brake practice know that the brake filling surge ought to be limited in order to avoid objectionable excess pressure (control chamber of variable brakes during release, and auxiliary reservoir in those without it). Generally, it is left to the driver to judge the extent of the brake filling surge in terms of the number of axles on his train, the make up of the train and the depression he has made when braking. As it is not easy to determine exactly the duration of the brake filling surge, the driver — prudently — nearly always makes it too short.

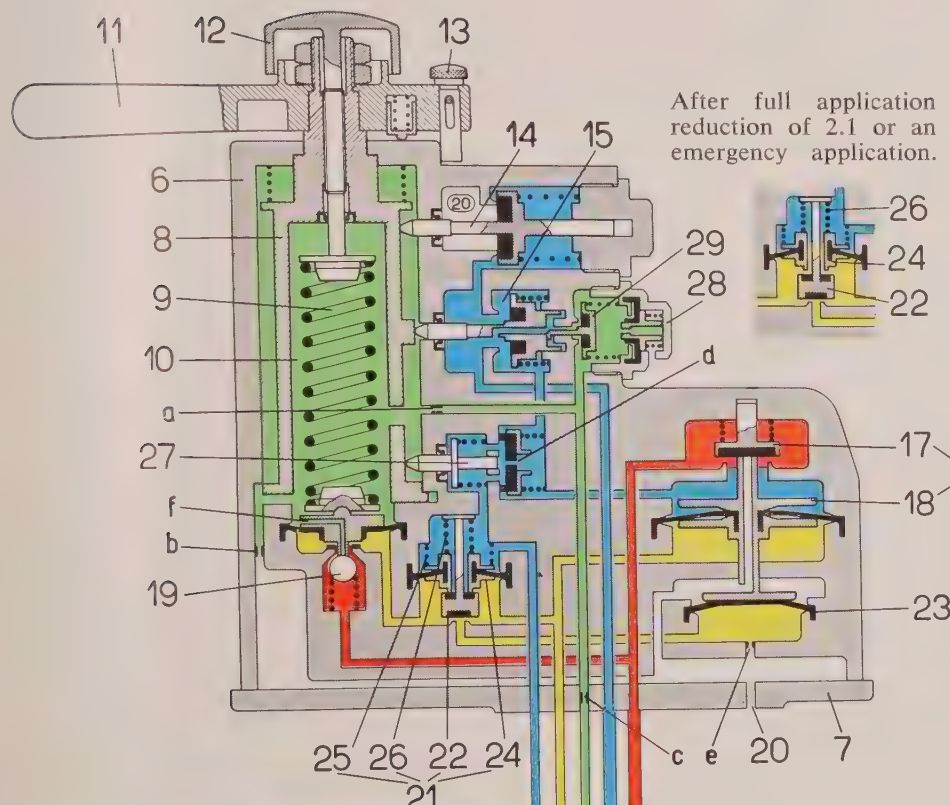
Modern designs of triple valves include a protective device against excess pressure: normally it is almost impossible to excess charge. Although we shall have to put up with the older distributors for a long time, there are already in service train

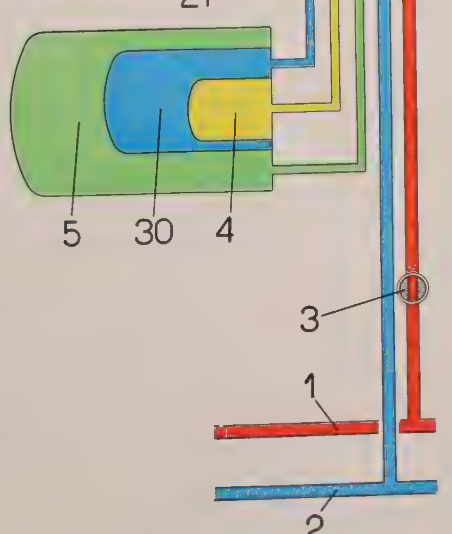
Oerlikon type FV 4 driver's brake valve.

Recharging position.

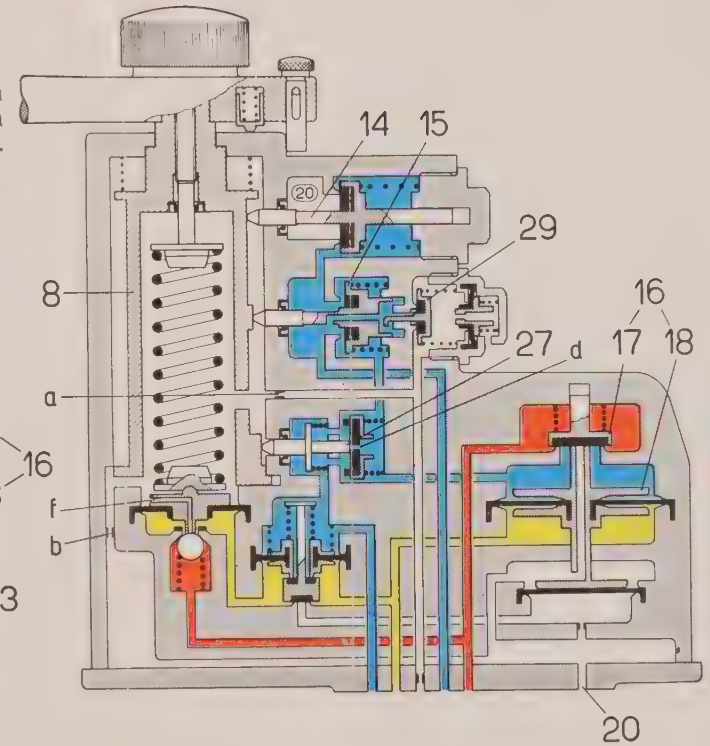
Release with automatic brake filling surge, after full application, reduction of 1.5.

After full application reduction of 2.1 or an emergency application.

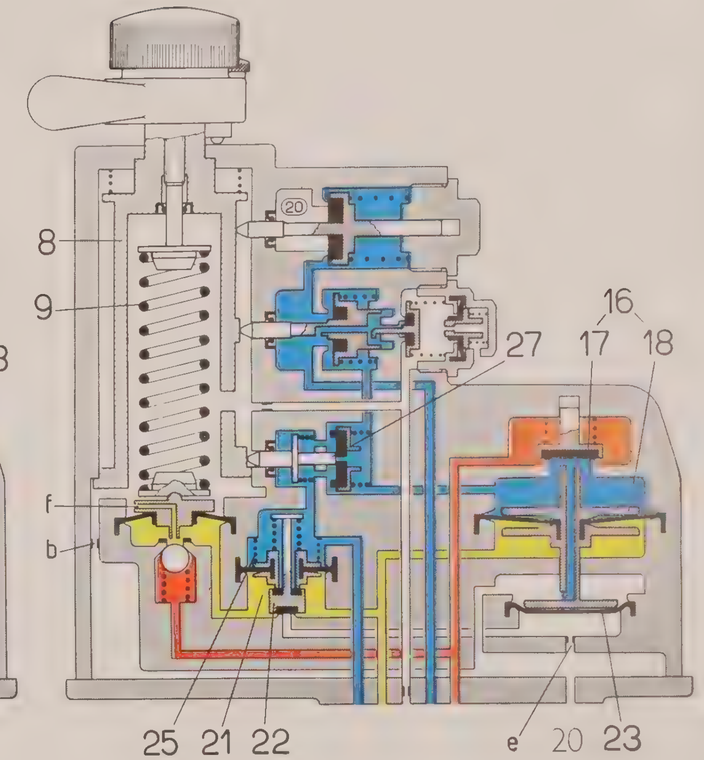




Running position.



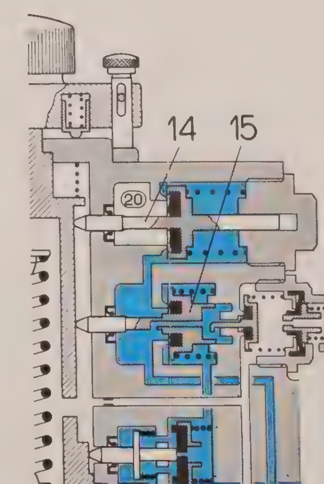
Normal position of application.



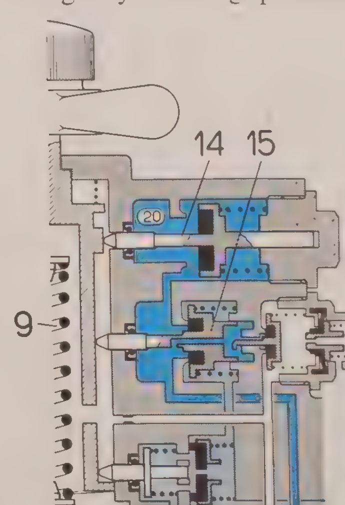
1. Feed pipe.
2. Train pipe.
3. B.V. valve.
4. 1 l. control reservoir.
5. Reducing reservoir, 7 l.
6. Body of valve.
7. Base with connection.
8. Operating sleeve.
9. Regulating spring.
10. Pressure regulator.
11. Driver's valve handle.
12. Regulating cap.
13. Lock.
14. Emergency valve.
15. Cut off valve.
16. Transmission member.
17. Admission valve.
18. Operating piston.
19. Regulator valve.
20. Exhaust orifice.
21. Arrangement for the brake filling surge.
22. Valve of brake filling surge.
23. Diaphragn of the brake filling surge.
24. Piston of the device.
25. Spring of piston 24.
26. Spring of valve 22.
27. By-pass valve.
28. Cap with exhaust valve.
29. Low pressure overcharging valve.
30. Time control reservoir 3 l.

a. b. c. d. e. f. calibrated orifices.

Neutral position.



Emergency braking position.



sets fitted with modern triple valves with which full advantage can be taken of the modern equipment. The driver's valve too has been improved to avoid these excess pressures or at least to render them harmless. We may cite the following possibilities :

- limitation of the maximum pressure of the brake filling surge;
- limitation of the duration of the brake filling surge;
- slow and progressive diminution of the over pressure by taking into account the limit of sensitivity of the triples.

The principle incorporated in the Oerlikon type FV 3 for example results from the fact that an over-pressure limited to 5.4 kg/cm^2 need cause no anxiety especially if it is subsequently reduced automatically slowly and progressively. A brake filling surge limited to 5.4 kg/cm^2 can and should be of longer duration than one at high pressure. In the Oerlikon type FV 4 for long trains, the brake filling surge is limited to 7 to 7.5 kg/cm^2.

The H 7/FVf 2 valve used on the S.N.C.F. shows a further variance. This valve has a first re-charging position I without limitation of high pressure and in addition a second re-charging position II with pressure limited to 5.4 kg/cm^2 and automatic and progressive reduction of the overcharge to 5 kg/cm^2. This is an interesting modification of an old design of driver's brake valve.

The duration of the brake filling surge at high pressure can be limited automatically. This is done by interrupting the surge as soon as a small timing reservoir of the driver's valve is filled. The time to refill this reservoir is adjustable : it is normally set so as to correspond to the auxiliary reservoir of a sensitive brake placed at the head of the train.

Opinions are divided as to the desirability of this automatic control such as already existed in the Knorr type C automatic valve and recently in the type FV 4 Oerlikon valve. Based on our experience we are in favour of automatic control. To obtain short release times on long trains after a full brake application, brake filling surges of more than ten seconds are required. It is however difficult to judge the exact period of a long brake filling surge. To appreciate this a watch should be taken and twenty seconds counted. A control pressure gauge could be provided which would indicate to the driver the moment he should restore his brake handle to the running position : this however distracts the driver's attention when he ought to be watching the line.

By automatic control in practice a shorter release time is obtained and in addition excess pressure is avoided.

The criticisms sometimes advanced against automatic limitation of the brake filling surge are the following :

(*a*) Owing to there still being old brake equipments sensitive to excess pressure the brake filling surge has to be limited to so short a time that the optima release times cannot be got with a train fitted throughout with modern triples capable of standing longer.

(*b*) The influence of the length of the train on the duration of the brake filling surge is very slight.

We feel the following comments should be made as regards this.

PARTICULARS OF A NUM

		Dri			
		Westinghouse No. 4	*Westinghouse No. H 7 modified by SNCF: FVf2*	*F.S.*	*Bozič*
Features	Year of manufacture	prior to 1900	prior to 1930 altered : 1953	about 1930	1929
	Operation by rotary valve or by valve	slide valve	slide valve	slide valve	valve
	Positions	recharging running neutral graduated application rapid application	recharging I recharging II running equalisation neutral graduated application rapid application	recharging I recharging II running neutral neutral braking graduated braking rapid braking	recharging running graduated app ation rapid applicatio
	Automatic determination of the pressure	—	partially in recharging position II	—	yes
	Automatic compensation of losses of pressure	in running position only, weak	in running position only, more heavily than at No. 4	recharged position II : heavy running position : weak neutral brake position : weak	in each positio
	Properties particularly related to the brake filling surge	unlimited	in recharging position II limited to 5.4 with automatic release (FVf2)	in recharging position I : unlimited. In recharging position II : limited to 5.0 with larger orifices than in the running position	unlimited

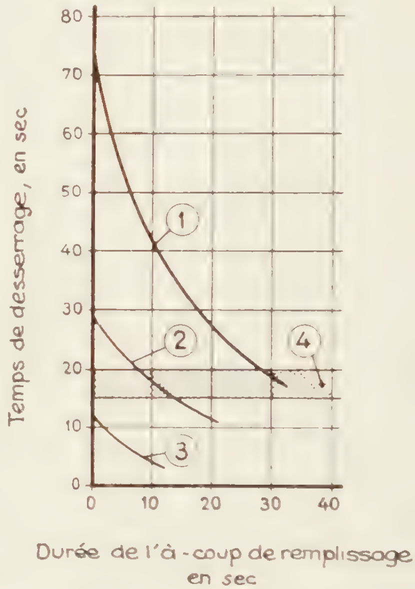
F DRIVER' S BRAKE VALVES.

ake valves				
Kradolfer-Weibel BLS	*Knorr-Automatic type C*	*Knorr-Automatic type D*	*Oerlikon FV3*	*Oerlikon FV4*
937	1935	1953	1950	1954
ide valve	Slide valve and valve	valve	valve	valve
echarging unning upplementary* eutral raduated application apid application with compensation f losses of pressure n the graduated application position	special double-headed running* graduated application rapid application * with latch in place of recharging position. In all positions gives a brake filling surge	recharging running average graduated application* rapid application * with 9 notches	double-headed recharging running First stage of braking 0.4* full application 1.5* full application 2.1* rapid application * depression in kg/cm²	double-headed recharging running First stage of braking 0.4* full application 1.5* full application 2.1* rapid application * depression in kg/cm²
——	yes	yes	yes	yes
unning position : eavy eutral brake position : weak	in each position	in each position	in each position	in each position
nlimited	automatically limited	unlimited automatic discharge of the over pressure	(overcharging at low pressure) automatic discharge of the overcharge	automatic limitation with overcharging — a low pressure followed by the brake filling surge and automatic discharge of the overcharge

Whereas with the old triples and driver's valves the duration of the brake filling surge should not exceed 10 seconds or so, this period can be almost doubled with a modern driver's valve — in conjunction with old designs of triples — thanks to the automatic and slow diminution of the over pressure which follows the brake filling surge. As the curve of figure 1 shows the longest passenger trains can with a brake filling surge of 20 seconds be released in less than 30 seconds. Even with long goods trains the 20 seconds brake filling surge is more than ample. Figure 2 shows that a train of 120 pairs of wheels after a full brake application can be released in 50 seconds which corresponds to the time of release of an isolated wagon and therefore represents an absolute minimum. With very long trains of 150 pairs of wheels, the time of release has been 70 to 80 seconds with one brake filling surge of 20 seconds. This release time can be further shortened by starting to diminish slowly the over-charging earlier from a higher pressure. In principle the duration of the automatic brake filling surge can be varied to suit the condition, as for example by a change of working conditions.

In the case of the Oerlikon FV 4, the automatic brake filling surge at high pressure is followed by the over-charge at low pressure of about 5.4 kg/cm2. This pressure is maintained as long as the handle is left in the first position. This is a useful method by which to extend the automatic brake filling surge, but at a lower pressure and in untoward circumstances less dangerous. On restoring the driver's handle to the running position, the over pressure is eliminated slowly and the working pressure restored.

In the Knorr D, the maker has abandoned the automatic limitation of the brake filling surge. He is satisfied with the automaticity of the processus of the diminution of the excess charge which by

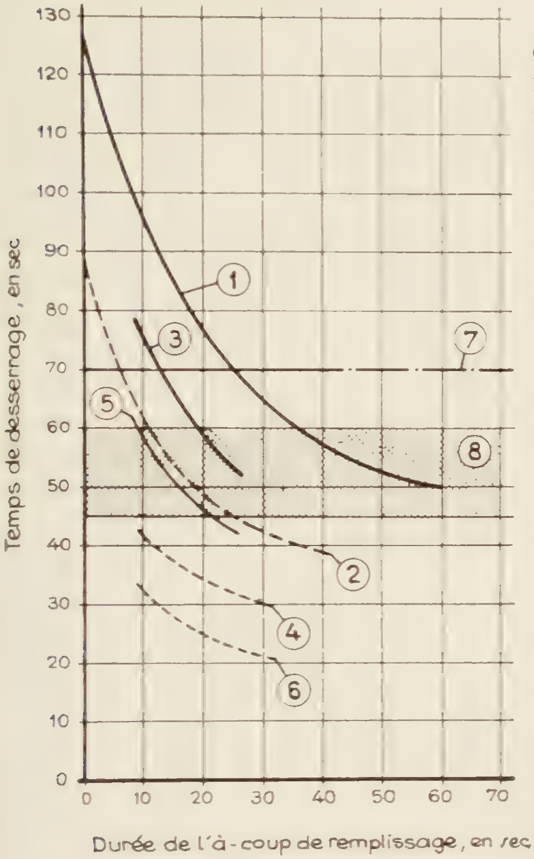


N. B. — Temps de desserrage en sec. = release time in sec. — Durée de l'à-coup de remplissage en sec. = duration of the brake filling surge in sec.

Fig. 1. — Release time with a rake of 80 pairs of wheels (passenger brakes) relatively to the duration of the recharging brake filling surge and the value of the braking.

(1) After full application.
(2) After graduated brake application 1 kg/cm2.
(3) After gratuated brake application 0.5 kg/cm2.
(4) Time of release of the wagon taken singly according to the I.R.U. (U.I.C.) requirement (15-20 sec).

itself is enough to avoid uncontrolled applications. This solution has the advantage of enabling longer brake filling surges to be given where conditions allow and moreover it provides means for increasing pressure where this could have drawbacks.



N. B. — Temps de desserrage en sec. = release time in sec. — Durée de l'à-coup de remplissage en sec. = duration of the brake filling surge in sec.

Fig. 2. — Time of release of goods trains relatively to the duration of the brake filling surge and the value of the braking.

(1) Rake of 150 pairs of wheels, three quarters braked after full application.

(2) Rake of 150 pairs of wheels, three quarters braked after graduated application at 1.0 kg/cm².

(3) Rake of 150 pairs of wheels, half braked, after full application.

(4) Rake of 150 pairs of wheels, half braked, after graduated application at 1.0 kg/cm².

(5) Rake of 120 pairs of wheels, three quarters braked, after full application.

(6) Rake of 120 pairs of wheels, three quarters braked, after graduated application of 1.0 kg/cm².

(7) Rake of 150 pairs of wheels, three quarters braked to I.R.U. (U.I.C.) condition (70 sec.)

(8) Single wagon to I.R.U. (U.I.C.) condition (45 to 60 sec).

The criticism according to which the duration of the brake filling surge adjusted for long trains could result in too long surges with short trains has shown itself to be unjustified by experience. It is true that the brakes at the head of a short train receive higher over pressure than on long trains but thanks to the slow automatic diminution of the over-pressure the brakes do not apply uncontrolled. This danger is even less great in short trains through the slower automatic diminution of the over-pressure.

(*c*) *Elimination of the over-pressure.*

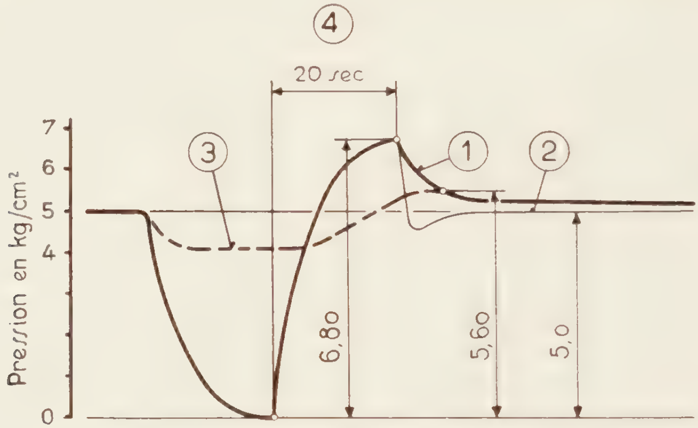
Let us suppose we have at the head of the train a brake with the reservoirs over-charged, that is to a pressure exceeding 5.0 kg/cm², the quick restoration of the driver's valve handle from the charging position to the running position will cause a fresh application.

To avoid this drawback, the driver brings back the brake handle slowly; this absorbs all his attention and prevents him from keeping his eye on the line. In spite of this he does not always succeed in preventing the brakes going on.

On this point the automatic action is a very great improvement. By a suitable choice of the calibrated orifices and of the volume at the driver's valve even a heavy overcharge of more than 0.5 kg/cm² is automatically eliminated without any drawbacks (fig. 3). Thanks to this fact, the brake filling surge at high pressure may be said to present no danger.

The curve of diminution of the over-charge has to be chosen in such a way that on the one hand the brakes release quickly — always without unintentional

fresh application — and on the other the overcharge is to be eliminated as quickly as possible seeing that during the period of overcharge a warning or alarm application at the rear of the train might not be noticed by the driver. In view of a fast release, the slowed down elimination of the overcharge ought to commence at a losses of pressure in the train pipe on cylinders through leakage to be made good automatically, that is to say not only in the running position of the driver's handle but also when it is in the brake applied position. I was felt to be necessary when designing the first driver's brake valves for use with brakes with



N. B. — Pression en kg/cm² = Pressure in kg/cm².

Fig. 3. — Release with brake filling surge limited automatically. Automatic elimination of the excess pressure in the train pipe and auxiliary reservoir without uncontrolled braking.

(1) Train pipe, diagram given by a modern valve.
(2) Train pipe, diagram obtained with an old valve.
(3) Auxiliary reservoir of an old Westinghouse brake placed at the head of a train.
(4) Duration of the brake filling surge.

high pressure, for example 6.0 kg/cm^2 and take place slowly whereas to suppress the overcharge quickly the slowed down elimination should commence only at the pressure of 5.5 kg/cm^2. A compromise has to be come to. In any event the feed after a brake filling surge should be at least high enough for the train pipe pressure on long trains to remain above the working pressure (fig. 3).

(*d*) *Compensation of air losses.*

As has been mentioned already, brakes which are graduable during release require graduable release having a neutral position with feed to arrange a second neutral position without feed as there was fear there would be uncontrolled release of the brakes not graduable during release. This fear has been found to be unjustified with the modern valves and thanks to the high sensitivity of the valves and diaphragms, the feed takes place absolutely free from surges and progressively. For this reason, an absolute neutral position need not be provided; on the other hand, this position can be retained for other purposes.

The re-feed ought to be limited so that a passenger's communication signal, a broken coupling or a leak due to a closed brake application valve will be noticed by the driver.

(*e*) *Graduability.*

As being an important advance, attention must be called to the precision graduation. As regards the technique of handling the brake, the ideal will be attained when, whether to keep the speed exactly on down grades as to stop with precision in dead-end stations, no method of braking requiring a special mastery should be prescribed and the driver will attain the greatest precision by making at will light applications or releases. Precision graduation above all facilitates the handling of the brakes on easy down gradients over which small braking efforts are needed. With an old driver's valve the driver in such circumstances must all the time keep on applying and totally releasing the brakes.

(*f*) *Summary.*

The development first of all of triple valves made a similar development of driver's valves necessary. The progress made up to date is encouraging. Thanks to a wide degree of automaticity, the manipulation by the driver is simplified and the false manœuvres are reduced. Excessive pressures above all — formerly so harmful — are eliminated, which suppresses the greatest obstacle to the rapid release of the brakes of long trains. To-day it is possible to arrive at a maximum result, that is to say, to release a long train in about the time required to release the brake on a single vehicle.

The new driver's valve is not only designed to suit the modern triple valves, but it offers at least as many advantages when it controls the older triple valves (for example the elimination of excess pressure).

The various modern designs of driver's valves do not differ very much in principle. As regards these differences, experience will show the best solution.

(*g*) *Present state of development.*

The problem of the development of the driver's brake valves occupies the attention of nearly all railway managements as that of all brake manufacturers. Moreover the I.R.U. (U.I.C.) Brake Sub-Committee is investigating the conditions and recommended requirements with which a modern driver's brake valve has to comply. Although the choice of a driver's valve may be a national matter, there are certain international obligatory requirements which require driver's valves to have some uniform features, for example, with a view to avoiding difficulties when changing locomotives at frontiers.

Furthermore, the future development of international traffic leads us to foresee that in the future, motor vehicles will pass the frontiers more and more frequently. Finally, there can be nothing but advantages for all railways to exchange their experiences in this field.

[625 .24]

The « Waggonfabrik Uerdingen » proposal for the wagon of the future,

by Dr.-Ing. E. h. Ernst KREISSIG, Krefeld-Uerdingen.

(*Eisenbahntechnische Rundschau,* No. 10, October 1956.)

I. Fundamental conditions for an economic method of building wagons.

In the I. R. U. (U. I. C.) competition for the design of an open high-sided goods wagon of the future, as a result of which the panel of judges awarded first prize to the proposals of the « Waggonfabrik Uerdingen », the directives given to the contestants provided in the first case for the design of economic types of wagon.

The economics of a type of wagon do not result only from a low cost of purchase, which causes the annual replacement charge to fall, nor from light weight construction which improves the operating results by reducing the cost of haulage. On the contrary, it is of fundamental importance that the costs of day-to-day maintenance and repairs should be taken into account, as they increase progressively with longer periods of use. They provide a definite limit to the economic life of the vehicle and it must be remembered that developments in technique and design cannot be applied to the oldest vehicles. On the basis of these considerations, the conditions of the competition called for a design which would provide a life of 20 to 25 years.

Moreover, an obvious preliminary condition for the most economic use possible of a future standard and common-user European stock of wagons, lies in the adoption as far as possible of standard or current parts, or at least those which have been widely and satisfactorily used in service for maintenance, repair and replacement and for which there are in all countries the most suitable technical conditions.

To resolve the problem set by the competition, it was necessary first of all to determine and to eliminate as far as possible all the features liable to increase the costs of maintenance and repairs, particularly normal wear in service, influence of the weather, corrosion, irregular shocks during shunting, damage caused during loading and unloading — particularly with automatic hoppers. The starting point for the investigation was a critical study of all parts of the wagon from the point of view of functional suitability, liability to corrosion and possibly fatigue, and the effects of these on the life of the parts. Special attention had to be given to providing the minimum weight for a determined stress, because this is where every reduction realised is beneficial to the load to be conveyed and at the same time reduces the proportion of traction costs attributable to the tare.

II. Precedents to the Uerdingen proposals.

Since 1930, the « Uerdingen Waggonfabrik » has been supplying to the Wanne-Herne Company a hopper wagon for the carriage of coal, with a diagonal flexible frame of the hollow beam type which has given such encouraging results from both operating and economic view points that these wagons have been widely used. In 1934, a large number of these vehicles, in the two-axle form, were supplied to the German State Railways; they were built with hollow beams of 6 mm wall thickness, with diagonal resilience, but manu-

factured for the first time in ST. 52 steel with an addition of 0.2 % copper for corrosion resistance. The resistance to corrosion of these vehicles, as well as their great resistance to plastic deformation led the Deutsche Reichsbahn to use this type of construction with hollow beams of ST. 52 steel enriched with copper for open goods wagons. In 1935, a fairly large number of these were put into service; it was, however, impossible to obtain details of their operating results because the outbreak of the second world war soon afterwards stopped all work of this kind. These wagons, when they could be recovered, were examined only after the war, but impressions were clearly favourable. The hollow beams were perfectly free from rust, both inside and outside; only where woodwork had covered the frame were there signs of an attack of rust in the form of scale, due to the addition of copper. Whilst in older types of wagons, there is at these points a stratified rusting which soon causes a rapid reduction in the thickness of the wall, the copper/steel beams were perfectly sound; this form of corrosion thus apparently protects the surface against an extension of the rusting. The same feature was also noted at other points. However, the particularly interesting feature was that after more than ten year's service, the repairs necessary to these wagons was remarkably small. These observations encouraged the Deutsche Bundesbahn to pursue with keeness the improvement of this type of construction, in collaboration with the Uerdingen Works. This work led to the D. B. open goods wagon of the type with hollow beams and complete diagonal flexibility (fig. 1); in this vehicle, not only is the frame fundamentally different from the usual rigid type, but the body itself is designed on up-to-date lines. The arrangement, up to now standard, of a door in the side has been retained.

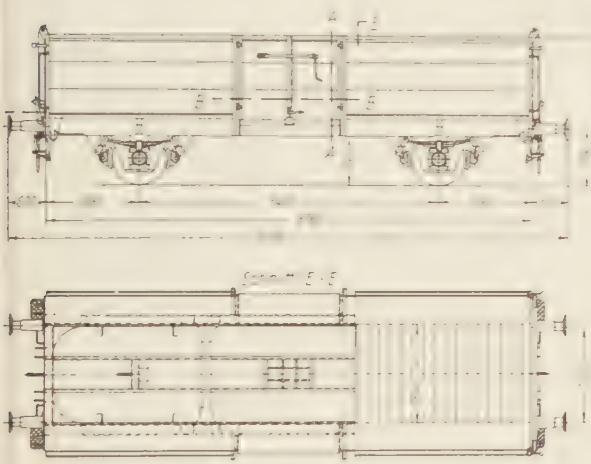


Fig. 1. — Open high-sided goods wagon of the German Federal Railways, with hollow frame beams and diagonal resilience.

Fig. 2. — Section of door sill on wagon in figure 1.

Figure 2 shows a section (A-A of fig. 1) of the ribbings (*c* and *f*) provided in the door plates. The same applied to the side walls. Figure 3 also shows the top streng-

thening of the door sill by means of hollow beams (*a*) and bottom stiffening a hollow beam (*c*) as well as the hollow pressing (*b*) designed to secure the door when closed; the same illustration also shows the vertical stiffeners for the door plates (section B-B). The plates of the end doors have the same arrangement (fig. 1).

The side wall units have been built on the same lines of ribbed stiffening; they outwards whatever the place or effort, which transfers the shock to the riser angle and thus to the door pillars. The latter are connected to the box-section solebar to which they impart a torsional stress. Like all hollow beams, especially those with walls of equal thickness, they have a very high capacity in relation to the free length of the stressed beam so that the energy is absorbed in a purely resilient way. It is

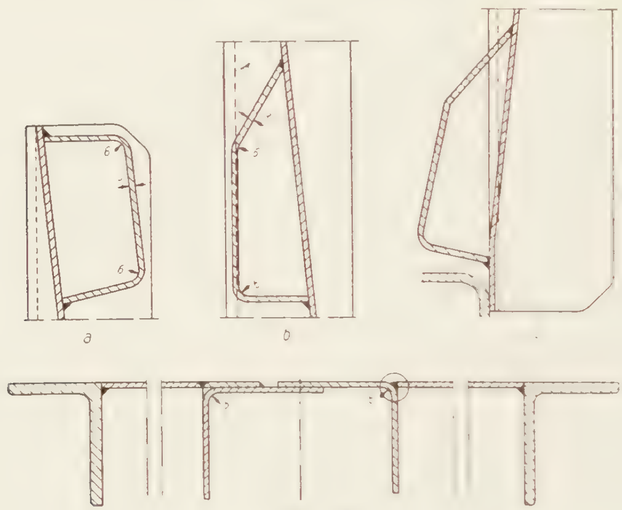


Fig. 3. — Arrangement of door on wagon in figure 1.
N. B. — Schnitt B-B. = section B-B.

are efficiently reinforced by door and angle riser by hollow stiffener beams along the top and at the bottom by floor plates which strengthen the walls and are connected to the hollow longitudinals (fig. 1). In this way, they are able to achieve satisfactory absorption of active and passive loads in an oblique direction and it was moreover possible to save four side stanchions per wagon. Another result equally desirable was the increased absorption of effort by these resilient side wall units, especially having regard to the risk of damage to the wall during unloading: if, for example, the grab of a crane strikes the top sill of the wall, the hollow beam forming the edge is subjected to deflection inwards or important only that the frame stays should be located as far as possible from the door pillar to ensure a volume of work sufficient for the solebar to absorb the energy.

Figure 1 shows the arrangement of frame stays, and solebars, which are in the form of box-beams. To allow fixing of the buffers and continuous drawgear, the headstocks are formed as open beams, but their moment and modulus of inertia correspond to those of the middle bearers. The drawgear is mounted flexibly between two centre longitudinals whose horizontal moment of inertia is calculated so that they do not prevent the resilient deformation of the frame.

The running gear is the standard

Deutsche Bundesbahn type, with the difference that the spring supports are of hollow beam shape.

The most important factor in reduction of maintenance cost, however, is the energy absorption capacity of the buffers, under the form of a hollow beam, and other cross-members to take the floor pressure, but the horizontal moments of inertia of these cross-members are so low that they do not in any way impair the diagonal resilience of the frame.

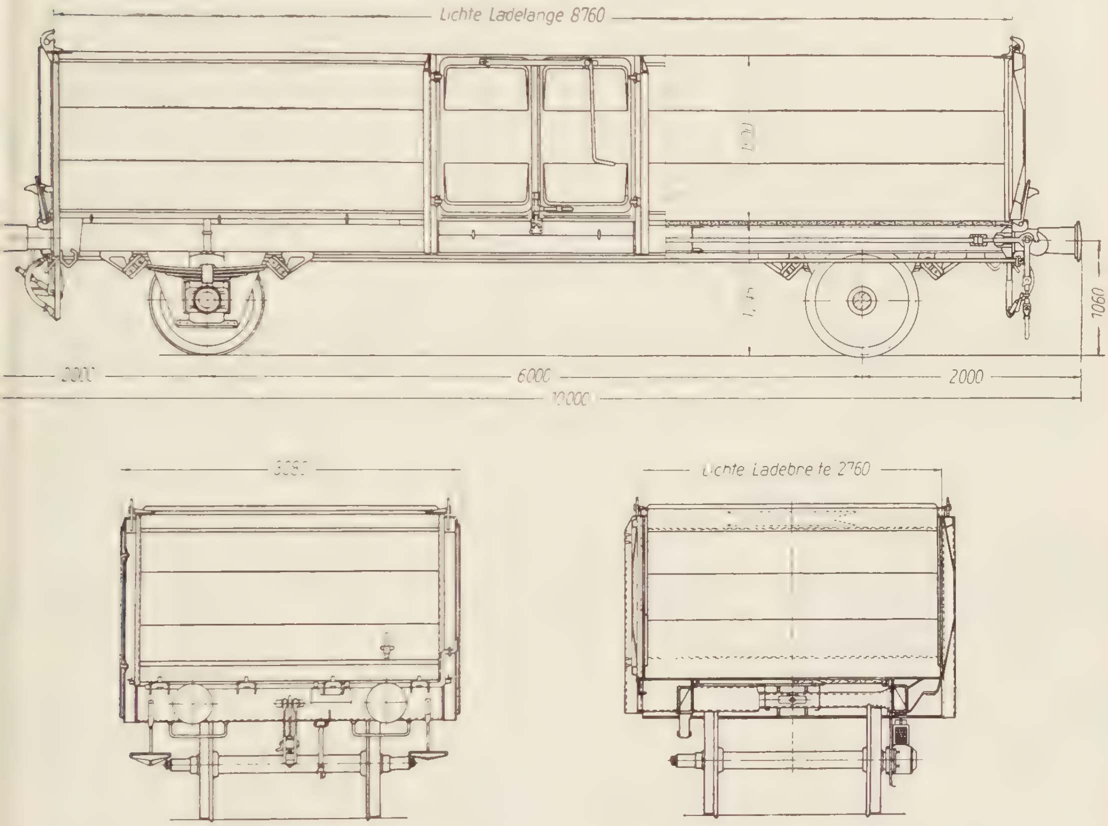


Fig. 4. — Wagon suggested as proposal No. 1 for the competition.

N. B. — Lichte Ladelänge = internal length for load. — Lichte Ladebreite = internal width for load.

the effect of axial forces as well as eccentric forces. This question has been dealt with in detail, along with the resilient frame is Nos. 5 and 11 of the *E. T. R.* (1952). The frame used as a basis for the calculations in these articles has, unlike the wagons of the Deutsche Bundesbahn (fig. 1) a single middle bearer in

III. The competition proposal.

The results so far obtained from these wagons led to the use of the basic components of their design in the proposals of the « Waggonfabrik Uerdingen » for a « wagon of the future », account being taken of latest technical progress. As this was a competition of ideas, three variations

of the same design were submitted for consideration by the panel of judges :

I. — A proposal with solebars of pressed steel in the form of rectangular hollow beams;

II. — A proposal with solebars of open angles;

III. — A proposal with tubular solebars.

been provided in pressed steel plate, 4 mm thick, with locking on the top edge. It is thus possible to use a special pressing to take the catch; moreover, the loading moments resulting from oblique pressures are logically absorbed by the upper and lower edges of the stiffened door. The provision of pressed steel doors thus saves weight and welding.

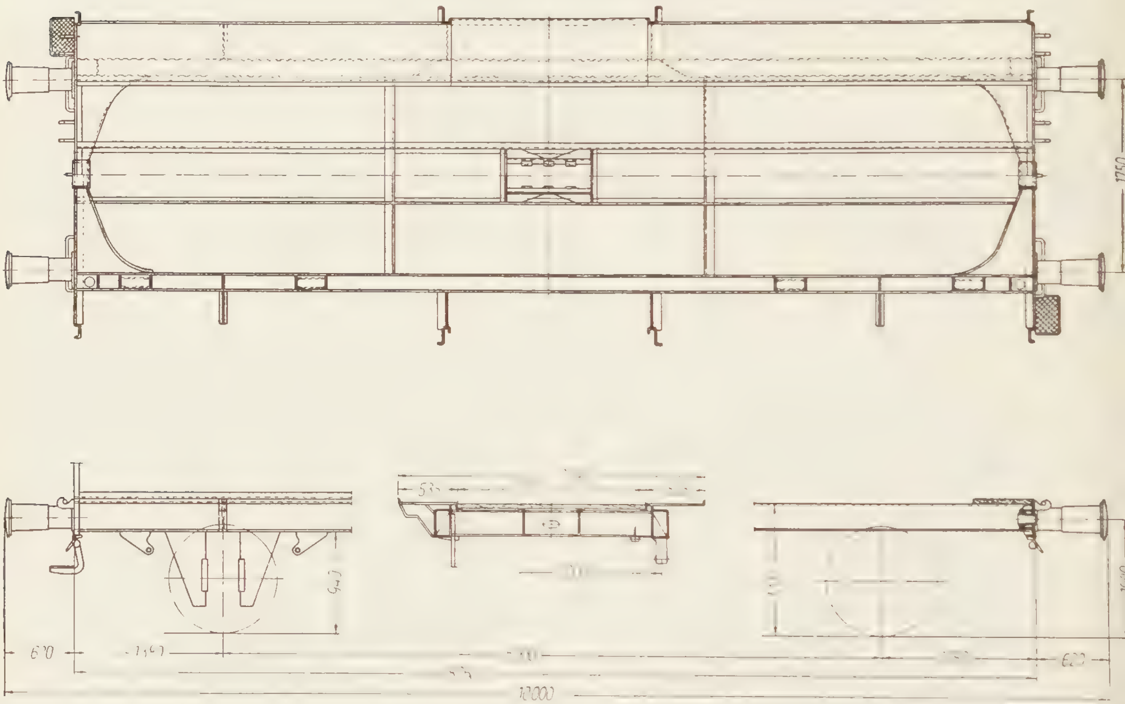


Fig. 5. — Frame of proposal No. 1.

a) *First proposal.*

Figure 4 gives end and side views as well as a section through the side walls of the wagon in proposal No. 1; figure 5 is the corresponding frame with various sections and views.

Figure 4 shows that the superstructure of the body corresponds generally to the Bundesbahn type 0.52 m; some alterations have been made to the design of the side doors and drop ends. The side doors have

The drop ends are also of 4 mm steel plate and can also be provided as stampings in the case of large quantities being required. For smaller quantities it is preferable to use pressings to avoid the high cost of dies. The mounting of the ends, as well as the cranked catch and safety lock are arranged in the normal way and comply with the I. R. U. (U. I. C.) regulations regarding interchangeability.

The solebars are, as shown in figure 5,

hollow and made up of 4 mm steel plate with a welded gusset, 5 mm thick. The full section is used for buffing forces, whether axial or not. The height of the section is such that it gives a beam with a high modulus of inertia, capable of absorbing, according to the existing conditions, the moments applied to the middle part of the wagon, i.e. in line with the door apertures, without the need to use stiffening plates or similar arrangements. The sections of the beams are produced by pressing and joined by automatic welding. The top gusset forms part of the floor and at the same time gives the beam a high transverse modulus of inertia, which serves to absorb the eccentric buffer loading of 40 tons prescribed by the conditions of the competition. This replies to the details of the diagonally rigid frame which does not have sufficient elasticity to absorb efforts due to eccentric shocks to the point where they are imperceptible. In the diagonally-resilient form, the diagonal elasticity obtained allows the unilateral load imposed on the frame to be limited under the action of eccentric shock to a value much less than the unilateral buffer load of 40 tons specified. Similarly, stresses caused by eccentric buffing forces can be reduced as desired.

The headstocks are provided as hollow sections formed by a buffer carrying beam and a pressed beam. An essential characteristic is the deep insert of this box type beam in the two solebars.

For the frame stays, use has been made of two U sections, 240 × 85 × 95. These serve in particular to support the bending moments resulting from side pressure in loading, transmitted by the door pillars. In addition, the middle longitudinals are effectively helped by these stays to absorb floor loading and all the longitudinals are kept at their normal spacing. The beams mentioned make up a frame of the most simple type, which is distinguished by its resilience under the action of diagonal buffing forces. With this type of frame, the maximum horizontal bending moments are produced at the points of connection. With the headstocks, and as the side faces of the solebars already have the necessary modulus of inertia and in addition the headstocks have a wide connection with the solebars, it is easily possible to meet the basic requirements of the competition for a unilateral buffer load of 40 tons.

Because of the elasticity of the frame, the constructional tolerances of the frame, buffer springs and cases, as well as buffer play in the case of a wagon oblique to the track, are *resiliently* compensated, so that in the event of heavy shocks, exceeding the capacity of the springs, the undamped shock after the initial blow is transmitted equally to the two solebars. In a frame of the *rigid* type it would, on the contrary, give rise to considerable stresses. The rigid frame would then distort *non-resiliently* to give a non-rectangular shape which would no longer be suitable or perfect guiding of the axles and would consequently cause excessive flange pressures on the rails.

One of the weaknesses of open goods wagons so far built is the fact that the low capacity of the buffer springs has already been exhausted at reduced speeds of impact — 8-12 km/h according to the type used — so that the frame is subject to extremely high stresses as a result of undeadened shock. Ring springs are in this case the most suitable resilient agent as within the reduced volume they allow not only an increased effort but also a longer working stroke, so that the solebars are not normally to the original maximum stresses which were the cause of frequent frame damage [1]. In the first proposal, and also in the second proposal, provision is made for a buffer power of 2 900 kgm making a final power of 48 tons with a working stroke of 120 mm and a buffer length of 620 mm.

The proposed arrangement thus provides protection for the vehicles against damage, from impact speeds of 15 km/h and more.

---

[1] The figures are included in an article on eccentric buffer forces, in Nos. 5 and 11, « E.T.R. », 1952.

The spring stroke of 120 mm does not, it is true, comply with the conditions of the competition, but the buffer can easily be replaced on the vehicle by the normal I. R. U. (U. I. C.) buffer by discarding the benefits of its greater power. For the continuous drawgear use has been made of the normal interchangeable units conforming to the I. R. U. (U. I. C.) regulations.

The wagon can, in accordance with the conditions of the competition, have a wheelbase of 5.4 m as well as double coil spring suspension. From operational trials on the German Federal Railways, however, the wheelbase has been fixed at 6 m and a single coil spring suspension fitted: this results in a reduced purchase price as well as equally good running qualities up to 75 km/h, as compared with the specified arrangement.

The axles are in accordance with I. R. U. (U. I. C.) requirements and have a tread diameter of 1 000 or 900 mm. Tests at present being carried out by the German Federal Railways will show whether, as some experts have suggested, it is possible further to reduce the wheel diameters. The wheels are designed on the monobloc system, centres and tyres thus being in one piece. Compensation for flange wear can

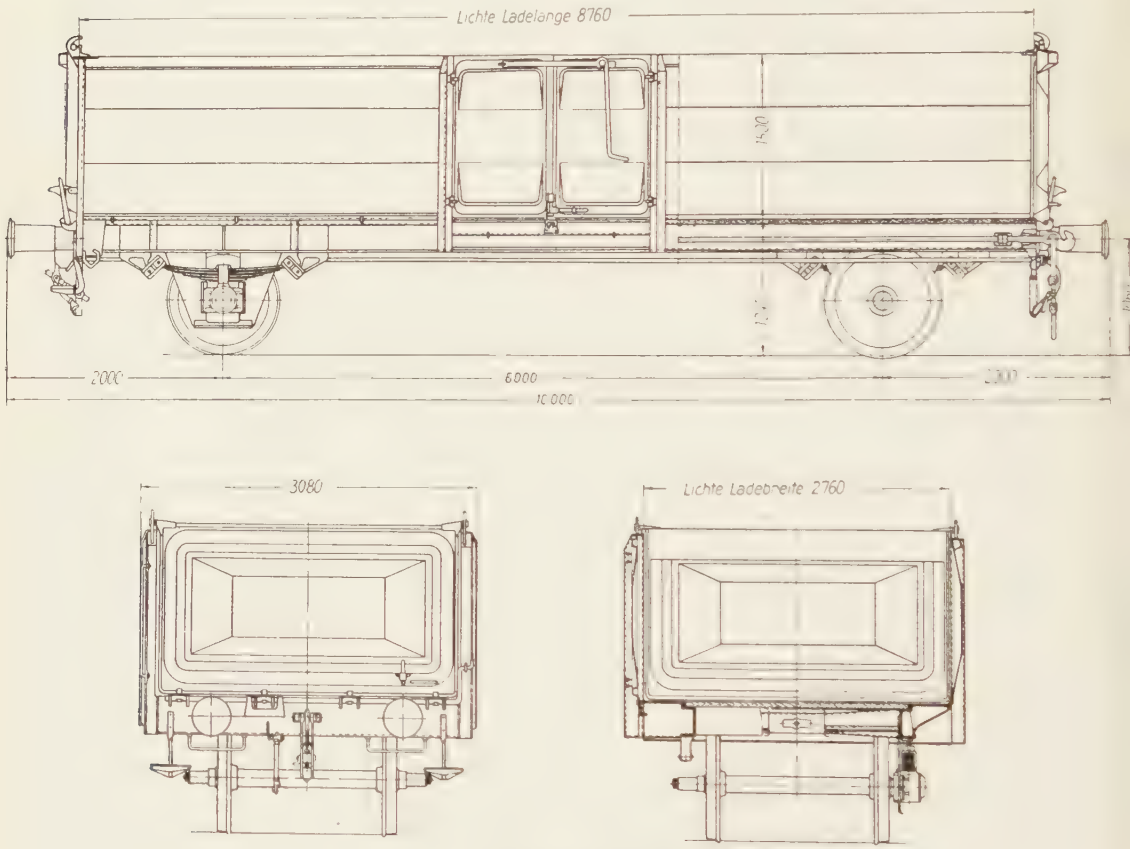


Fig. 6. — Competition proposal No. 2.

usefully be done by welding up followed by machining; this process has over a number of years shown its economic advantage for correction of flange profiles, as purely mechanical re-turning of the running surface means that side wear of the flanges necessitates the removal of a large amount of valuable metal in a radial direction, so that the removal of metal to restore the tyre profile reduces the useful life of the tyres much more than the wear itself.

The roller bearings and laminated springs are in conformity with the I.R.U. (U.I.C.) regulations or with the specification for type 2 of the present standard open goods wagons of the O. R. E. The axleguards are formed from flat steel plate or wide iron flats. They are welded to the solebars along their upper edge and the two sides. Slots in the top part provide for the necessary lengths of welding. Below the bottom of the solebars, the axleguards are provided with the usual stirrup irons.

The middle part of the floor is of planks 45 mm thick, whilst the sides are formed by the solebar gussets. Whilst leaving sufficient area for nailing to stow miscellaneous freight, this arrangement provides reduced tare weight and less likelihood of damage. The spacing of the frame members to support the floor is designed so that loads arranged in accordance with I. R. U. (U. I. C.) regulations are positively carried. Because of their short length, each plank is fixed by two screwnails.

For protection against corrosion, all surfaces supporting the wooden floor on the longitudinals have, in addition to the specified paintwork, an intermediate isolating layer of bitumen-impregnated material. The floorboards — when they are of resinous wood — are impregnated in heat with a special carbonyle to protect them against chemical action from the freight carried and against the weather. Above the wheels,

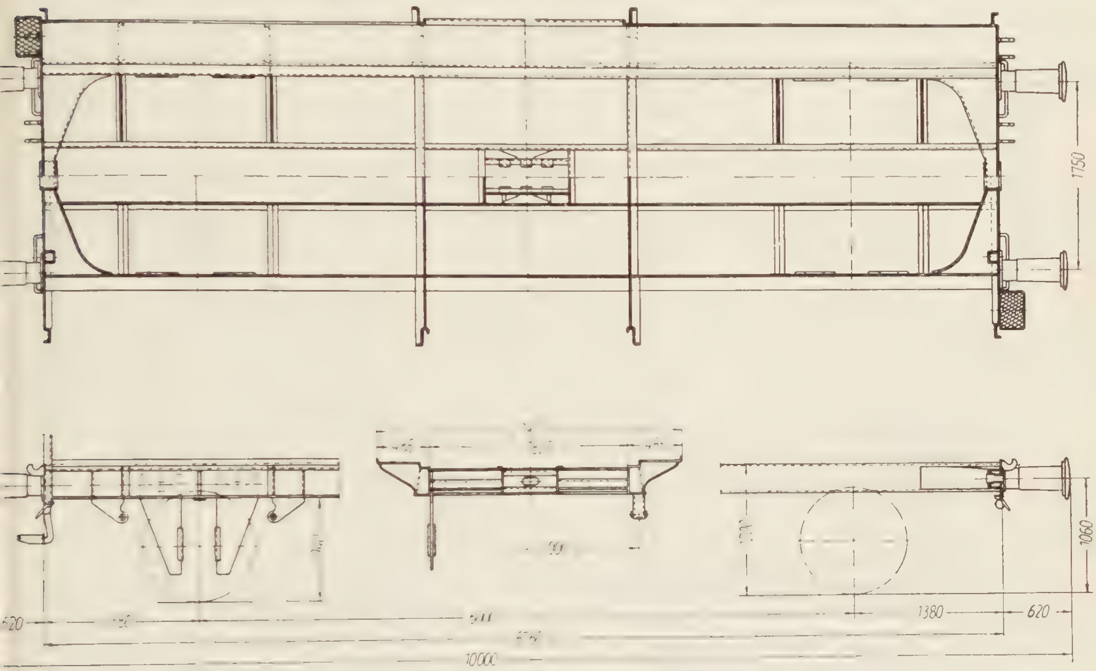


Fig. 7. — Frame of proposal No. 2.

they are coated with a special paint to provide protection against sparks.

Figure 1 shows a pipe-fitted wagon. Brake-fitted wagons are provided, in accordance with I. R. U. (U. I. C.) regulations, with compressed air brakes for example the KE 1 (*g*) with automatic slack adjustment. The mechanical part of the eight shoe brake is similar to that of the O. R. E. type 2 standard wagon. Brake fitted wagons to proposal No. 1, with oak floor, hand brake and enclosed brakesman's caboose, will have a weight of 9 700 kg with 1 000 mm diameter wheels and 9 100 kg with 900 mm wheels.

b) *Second proposal.*

Figure 6 shows proposal No. 2, a variation in which the solebars of the diagonally resilient frame are in the form of open beams of 7 mm steel plate, made up of pressings. It is intended to show that it is possible to build frames which are resilient diagonally, using open beams. The body of the wagon is the same as in the first proposal.

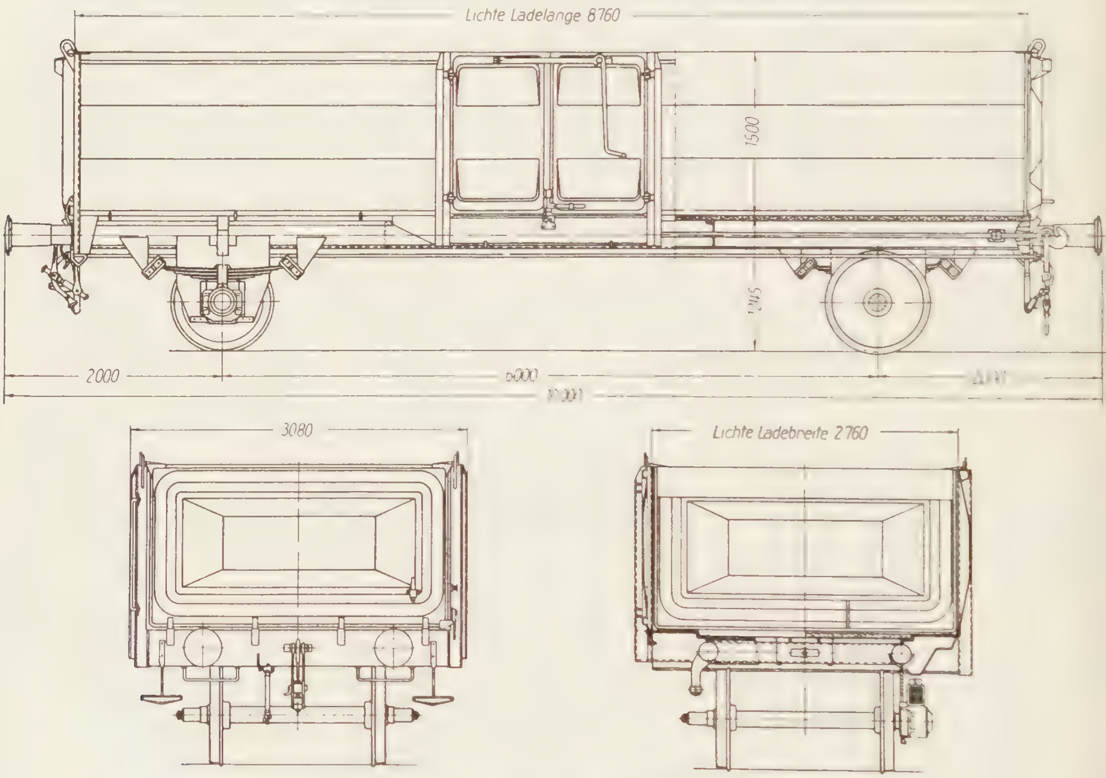


Fig. 8. — Competition proposal No. 3.

The frame (fig. 7) differs from that of the first proposal, not only in the difference in solebar section, but also in the positioning of the stays which are arranged in line with the door pillars and form with them a half portal. The stays and door pillars are bracketed together to provide the necessary rigidity.

With the same equipment as proposal No. 1 braked wagons should weigh 100 kg less.

c) *Third proposal.*

The third proposal (fig. 8) is another variation of the solebars, in the form of a closed tubular section.

The body differs from the two designs already described in the fact that the two end walls of the braked wagon are pivoted, one of them being equipped as a brake position. This arrangement avoids supplementary operations because the wagon can the friction of the oblique surfaces so that the end door opens automatically at a tipping angle of about 35°. When the wagon is returned to a horizontal position, the door is first moved by a handgrip *e* in the direction of opening and closed by thrust in the opposite direction, provided by the spring of the door itself which returns to the closed position by sliding over the external oblique surfaces of the retaining catch *b*.

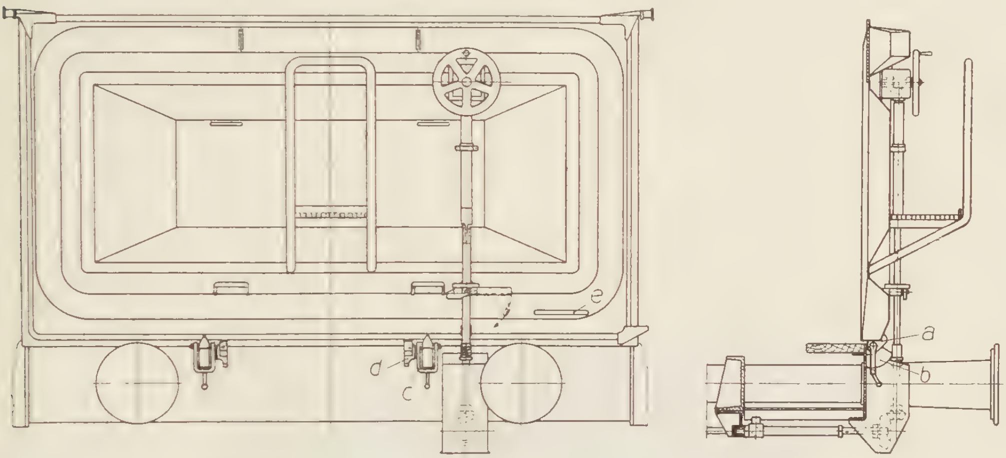


Fig. 9. — Arrangement for securing pivoted end door of wagon in proposal No. 3.

be run to an unloading point from either side. The brake screw, which retracts with the end wall, must first be lifted to release it from the conical pinions controlling the brake.

Another feature is the design of the catches for the pivoted ends. This is characterised (fig. 9) by the fact that the oblique surfaces *a* of the end wall bear against corresponding oblique surfaces of a securing bracket *b* on the headstock. The end door being protected against upward force by a safety stirrup *c* designed to prevent it being opened in the event of heavy impact. When the wagon is tipped longitudinally the locking stirrup, after manipulation of the safety lock *d,* allows the door to open and it is held in position during unloading by the same safety lock. The thrust of the load then overcomes

The frame solebars (fig. 10) are made up of tubes welded or solid drawn, 170-182 mm diameter with steel plate gussets 5 mm thick, welded on the crest. The section is fully utilised for axial and eccentric buffing forces. The height of the section gives a beam with a high modulus of inertia, capable of absorbing in the middle part of the wagon, in line with the door panels, the moments which can be produced under present conditions without necessity to use stiffeners or similar reinforcement. The top gussets form part of the floor and at the same time serve to give the beam a high transverse modulus of inertia, which is necessary to absorb a high specified test load acting on diagonally opposed buffers.

With regard to the headstocks and frame stays, it is sufficient to say that they have

been dealt with under proposal No. 1. The ends of the solebars penetrate the headstocks and form the buffer cases. They thus allow the power of the buffers to be increased as required, insofar as it is not restricted by the « Berne provisions ». In the present case the maximum stroke is 140 mm, which gives a maximum power of 2 300 kgm per buffer; this can naturally be further increased by enlarging the tube diameter.

The brake-fitted wagons, with oak floor, and wheels 900 mm diameter on tread, designed to proposal No. 3, with hand brake and brakeman's caboose, have a tare weight of 9 100 kg and with the open brake position and pivoted door, 8 900 kg.

d) *Features common to the three proposals.*

The new features described in sections *a* to *c* such as the high power of the buffers, for example, the end door locking, the open brake position on the end doors, are not restricted to any one of the proposals, but are applicable to all.

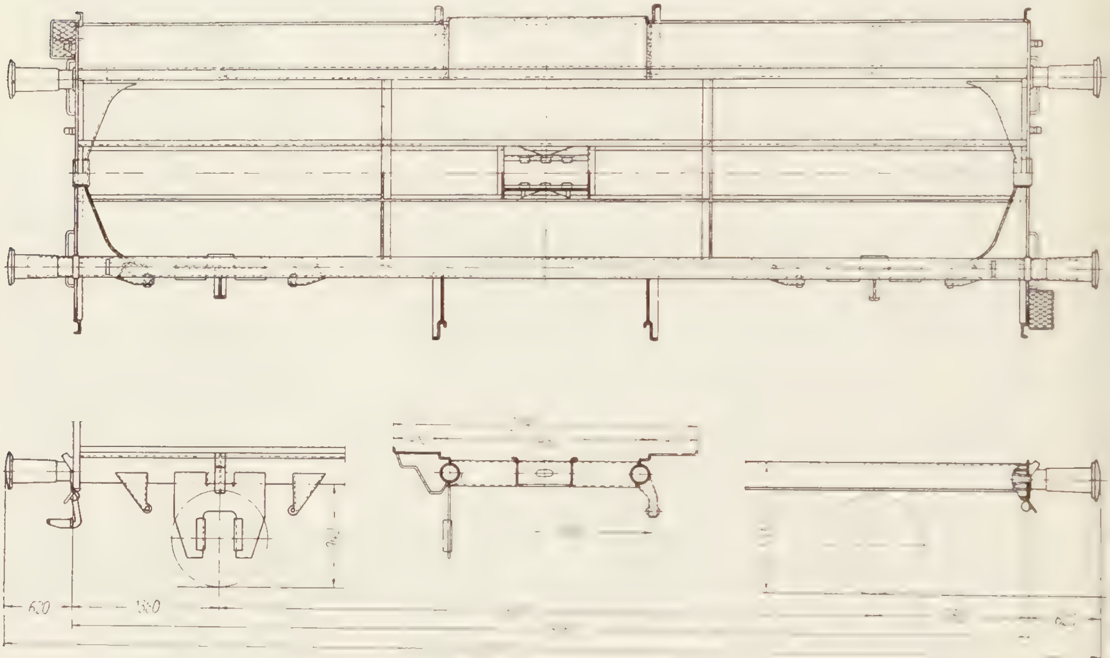


Fig. 10. — Frame of proposal No. 3.

The main dimensions of the wagon are the same for the three proposals :

Maximum length over buffers, without caboose	10 000 mm
Maximum length over buffers, with caboose	10 500 mm
Wheelbase as required	6 000 mm or 5 400 mm
Maximum width	3 080 mm
Maximum height	2 865 mm
Diameter of wheels	1 000 mm or 900 mm
Floor area	24 m²
Load space	36 m³

Needless to say, these dimensions apply only to existing vehicles; they will be modified as necessary in the light of the investigations into wheel diameters.

[625 .213]

Vibration tests to determine the most favourable ratio of axle springing and bolster springing with Minden-Deutz bogies,

by Doct.-Engineer Emil SPERLING and Dipl.-Engineer August POLAK, Minden (Westphalia)

(*Eisenbahntechnische Rundschau*, No. 11, November 1956.)

In order to determine the most favourable ratio between axle springs and bolster springs in the "Minden-Deutz" bogie, the German Federal Railways' research office at Minden have made tests on a roller stand. These tests and the results obtained are described. They are considered of great value for further improvements of the design of this bogie.

So far, there had been no clarification of the question of the most favourable ratio of axle and bolster springing. The French journal "Revue Universelle des Mines" (No. 8, 1952) contained an article by J.M. DEHALU about the "Amortissement des systèmes à double suspension et son influence sur le confort des véhicules". (Damping of systems with dual springing and its influence on the running quality of vehicles). According to DEHALU's theoretical investigations it would be more advantageous for the running quality of vehicles if the primary springing (axle springing) is softer and the secondary springing (bolster springing) harder. In contrast, the inverse ratio is applied on the German Federal Railways.

In order to resolve this question, roller stand tests have been carried out with a number of springing systems with the same overall springing effect but with different hardness ratio of axle and bolster springing. At the same time, an attempt was made to determine the most favourable ratio of primary and secondary springing.

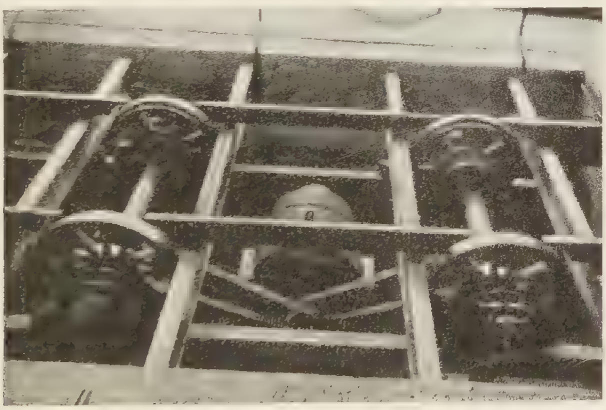


Fig. 1. — Roller test stand.

Roller test stand.

The roller test stand (fig. 1) on which the vehicles are exposed to stresses similar to those encountered in actual service, consists of two axles which can be moved horizontally in the direction of the rail the amplitudes of the vertical vibrations can be modified by shifting the appliances. The speed of the driving motor can be steplessly regulated so that the bogie of the stationary vehicle can be made to vibrate at frequencies of up to 11 cycles pr. second.

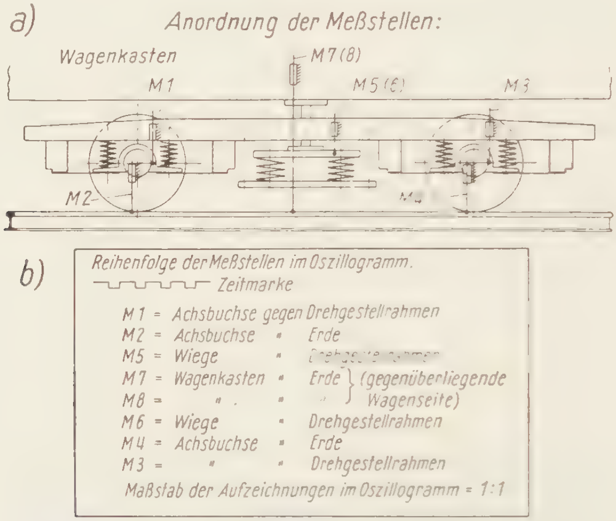


Fig. 2. — *a*) Arrangement of the measuring points.
b) Sequence of measuring points on the oscillograph.

N. B. — Wagenkasten = car body.
Zeitmarke = time mark;
M 1 = axle box against bogie frame;
M 2 = axle box against ground;
M 5 = bolster against bogie frame;
M 7 = car body against ground } (on opposite side)
M 8 = car body against ground }
M 6 = bolster against bogie frame;
M 4 = axle box against ground;
M 3 = axle box against bogie frame.
(Scale 1 in 1) :

and fitted with wheels. The tyre profile of the wheels corresponds to the profile of the rail head. The axles are driven by a Leonard-connected 40 kW motor by means of two Duplex chains. By means of different appliances on the wheels, it is possible to generate either vertical vibration alone, or horizontal and vertical vibrations simultaneously. Due to a slight eccentricity of appliances and wheel,

As the tests were exclusively concerned with the determination of purely vertical springing characteristics the only appliances used were those which cause vertical deviation. They were so adjusted as to cause a mean amplitude of excitation of 5.65 mm. In view of the fact that, in actual operation, the two axles of a bogie pass over a track irregularity with a certain time lag, the eccentricities of the test

stand axles were so adjusted that the amplitudes showed a phase displacement of 90° at the most.

Arrangement of the test.

The movements of the car body and the spring travels were recorded by means of a loop oscillograph. The displacements were measured by means of vertical displacement meters (resistance measuring bridges) in Wheatstone bridge connection.

Frame of the vehicle and bogie.

The tests were carried out with an old passenger coach which was due to be scrapped and which was fitted with Minden-Deutz bogies :

Weight of car body		25 780 kg.
Length over buffers		19 780 mm.
Length of car body	$L =$	18 480 mm.
Distance between bogie centres .	$a =$	13 250 mm.
Width of car body	$B =$	3 000 mm.

Fig. 3. — Test stand arrangement.

For these tests, altogether eight measuring points were chosen whose positions at the car body and at the bogie are shown in figure 2. Figures 3 and 4 show the complete arrangement of the test stand and the details of the measuring arrangements.

The different phases, at progressively increasing speed of the test stand axles, were recorded by means of oscillographs. The initial rotating speed of the axle was 30 r.p.m., corresponding to a frequency of 1/2 cycle pr. second. The speed was then increased to 40, 50, 60 and 65 r.p.m., thence by stages of 5 r.p.m. to 120 r.p.m. (corresponding to 2 cycles pr. second), and then by stages of 25 or 30 r.p.m. to about 660 r.p.m., thus covering the range from 2 to about 11 cycles in stages of about 1/2 cycle pr. second.

The moment of inertia of the car body in respect of the centre of gravity I_z, was approximately calculated from the following equation :

$$I_z \approx m \cdot \frac{L^2 + B^2}{12} \quad [\text{kg cm s}^2],$$

where : L = length of car body
B = width of car body
m = mass of car body.

Introducing the numerical values, one obtains :

$$I_z = 7.74 \times 10^6 \quad [\text{kg cm s}^2].$$

The moment of inertia in respect of a bogie pin, J_1, is obtained from the equation

$$J_1 = J_2 + m\left(\frac{a}{2}\right)^2.$$

where a is the distance between the bogie pin centres.

Introduction of the numerical values yields :

$$J_1 = 19.39 \times 10^6 \text{ [kg cm s}^2\text{]}.$$

The reduced mass m_2 of the car body, acting at the other bogie pin as application point, is obtained from the equation :

$$m_2 = \frac{J_1}{a^2} = \frac{19.39 \times 10^6}{1.76 \times 10^6} = \text{approx. } 11 \text{ [kg s}^2\text{/cm]}.$$

The vehicle was lifted by means of jacks at one bogie pin so that the vibrations of the springing of the non-driven bogie should not have a disturbing influence on the tests.

In order to enable the tests with the driven Minden-Deutz bogie to be carried out with simple as well as with double axle box dampers, brackets were welded to the bogie frame which were supported above the axle box, and each axle box cover was fitted with two additional lugs. Due to this arrangement, the additional dampers came to lie, in the same way as the single dampers, in a vertical plane defined by the centre line of the axle, and could be mounted and dismantled in a minimum of time. Figures 5 and 6 illustrate this arrangement with one and two axle box dampers, respectively.

The weight of the bogie amounted to 5 400 kg. After deducting the weight of the axles together with that of the axle journals, and of the brake blocks which had been removed to facilitate the tests, representing a total weight of 1 850 kg, the net weight of the sprung part of the bogie amounted to 3 550 kg. In consequence, the mass m_1 of the bogie is calculated as follows :

$$m_1 = G \cdot g = \frac{3\,550}{981} = 3.6 \quad \text{[kg s}^2\text{/cm]}.$$

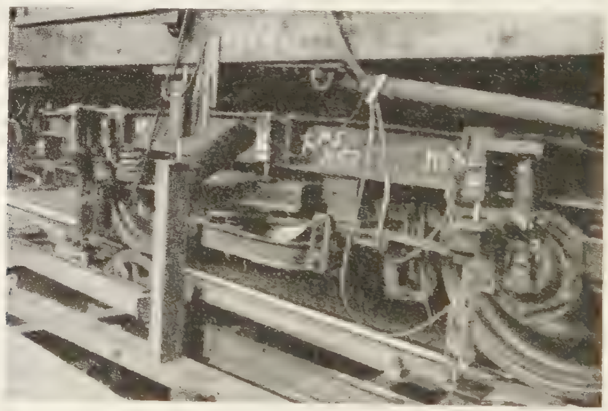


Fig. 4. — Measuring arrangements.

Spring and shock absorbers.

The tests were carried out with helical springs made of round steel, normally used for Minden-Deutz bogies, in the axle spring and bolster spring combinations tabulated in Table 1.

In the case of tests Nos. 4 and 5, the axle was sprung by means of bolster centre springs in order to obtain the necessary softness of the axle springing which it

TABLE 1. - **Combinations of axle springs and bolster springs in the tests.**

Test	Number of shock absorbers installed	Total springing Cg (cm/ton)	Ratio of axle and bolster springing
1	*a*) none *b*) one *c*) two	1.42	19.2 : 80.8
2	*a*) one *b*) two	1.43	30.7 : 69.3
3	*a*) one *b*) two	1.36	36.4 : 63.6
4	*a*) one *b*) two	1.46	66.5 : 33.5
5	*a*) one *b*) two	1.43	78 : 22

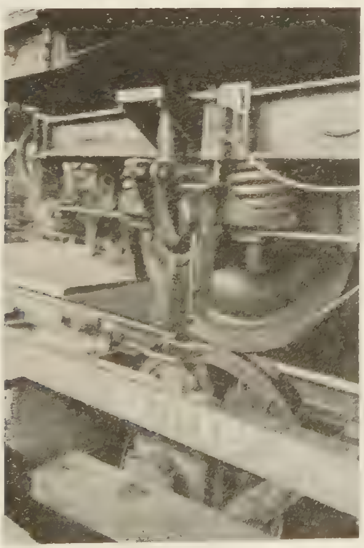
was no longer possible to obtain with normal axle springs.

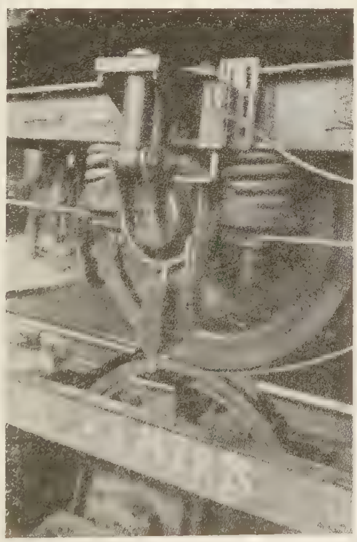
Prior to the tests, the constants of all the springs used were checked on a spring testing machine. It was found that their characteristics were in satisfactory agreement with the calculated values.

For the *normal axle damping*, consisting of one damper for each axle box, dampers of type T 40 × 25/70 were used. The mean damping force of these shock absorbers was determined by means of tests at a vibrating speed of $v = 10$ cm.pr.sec., and was found to be 262.5 kg. Their damping resistance is therefore :

$$\rho_{1_I} = \frac{4 \times 262.5}{10} = 105.0 \quad [\text{kg s/cm}].$$

The additional damping of the axles was obtained by four shock absorbers whose mean damping force was found by tests to amount to 492 kg at a vibration speed of $v = 10$ cm/sec. The additional damping resistance is thus :





Figs. 5 and 6. — Arrangement with one and two axle box dampers, respectively.

$$\rho_{1\,zus} = \frac{4 \times 492}{10} = 196.8 \quad [\text{kg s/cm}].$$

In the case of the tests carried out with double axle box dampers, one thus obtains a total damping resistance :

$$\rho_{1\text{II}} = \rho_{1\text{I}} + \rho_{1zus} = 105.0 + 196.8 = 301.8 \quad [\text{kg s/cm}].$$

For the *bolster damping*, dampers of type Ta 68 × 48/130 were used. The mean damping force of both shock absorbers was found to be 1 350 kg at a vibrating speed of $v = 10$ cm/sec. The damping resistance is therefore :

$$\rho'_2 = \frac{2 \times 1\,350}{10} = 270.0 \quad [\text{kg s/cm}].$$

Since, for the purposes of these tests, it is only the vertical component of the damping resistance ρ'_2 which is taken into account, this component was calculated from the dimensions of the damper suspension. The inclination of the dampers was $\alpha = 34.4^{\circ}$ so that the vertical component of the damping resistance ρ'_2 becomes :

$$\rho_2 = \rho'_2 \,(1 - \cos. 34.4^{\circ}) = 47.25 \quad [\text{kg s/cm}].$$

Calculation of the natural frequencies.

The natural frequencies of the car body, f_{o2}, and of the bogie frame, f_{o1}, were calculated from the following formula :

$$f_{o_{1,2}} = \frac{1}{2\,\pi} \sqrt{\frac{\nu_{1/2}^2 + \nu_2^2}{2} \mp \sqrt{\frac{1}{4}\,(\nu_{1/2}^2 - \nu_2^2)^2 + \frac{c_2^2}{m_1\, m_2}}};$$

where : $\nu_{1/2}^2 = \dfrac{c_1 + c_2}{m_1}, \quad \nu_2^2 = \dfrac{c_2}{m_2},$

c_1 = spring constant of the axle spring, c_2 = spring constant of the bolster spring, m_2 = reduced mass of car body, and m_1 = mass of the sprung part of the bogie.

The values of c_1 and c_2, expressed in kg/cm, for axle and bolster springs respectively, which applied to the tests with different spring ratios, are listed in Table 2. The values m_1 and m_2 remained constant for all the tests.

According to DEHALU, the forced amplitudes of the car body were calculated from the following formula :

$$\left(\frac{x^2}{y}\right)^2 = \frac{(1 + 4\,D_2^2\,\lambda_2^2)\,(1 + D_1^2 \frac{1}{\mu\varphi}\,\lambda_2^2)}{\left[(\lambda_{o_1}^2 - 1)\,(\lambda_{o_2}^2 - 1) - 4\,D_1 D_2 \lambda_1 \lambda_2^2\right]^2 + 4\,\lambda_2^2 \left\{D_1^2\,(1 - \lambda_2^2)\,\lambda_1 + D_2 \left[1 - \frac{1}{\mu}\left(\frac{1}{\varphi} + 1\right)\lambda_2^2\right]\right.}$$

where :

x_2 = amplitude of car body,

y = amplitude of excitation,

D_1 = attenuation constant of the axle dampers,

D_2 = attenuation constant of bolster dampers.

The attenuation constants are calculated from the formulas :

$$D_1 = \frac{\rho_1}{2\,m_1\,\nu_1}; \qquad D_2 = \frac{\rho_2}{2\,m_2\,\nu_2}.$$

where :

ρ_1 = damping resistance of the axle dampers,

ρ_2 = damping resistance of the bolster dampers,

TABLE 2. - **Spring constants of the primary and secondary suspensions natural frequencies of the car body and of the bogie frame, reduced mass of car body and of the spring parts of the bogie.**

Test No.	c_1	c_2	f_{o1}	f_{o2}	m_1	m_2
	(kg cm)		(cycles pr.sec.)		(kg sec² cm⁻¹)	
1	3 650	870	5.68	1.27	3.6	11.0
2	2 270	1 010	4.89	1.25	3.6	11.0
3	2 020	1 157	4.83	1.30	3.6	11.0
4	1 030	2 040	5.00	1.17	3.6	11.0
5	900	3 125	5.86	1.12	3.6	11.0

$$\nu_1 = \sqrt{\frac{c_1}{m_1}}; \qquad \nu_2 = \sqrt{\frac{c_2}{m_2}}.$$

The λ values are obtained from the following relations :

$$\lambda_1 = \frac{\nu_2}{\nu_1}; \ \lambda_2 = \frac{\omega}{\nu_2}; \ \lambda_{o_1} = \frac{\omega}{\nu_{o_1}}; \ \lambda_{o_2} = \frac{\omega}{\nu_{o_2}}.$$

where :

ω = speed of excitation $= 2\pi f_e$ (f_e = frequency of excitation)

$$\nu_{o_{1,2}} = f_{o_{1,2}} \cdot 2\pi.$$

The values for μ and φ are :

$$\mu = \frac{c_1}{c_2}; \qquad \varphi = \frac{m_2}{m_1}.$$

The resonance curves calculated for $\left(\frac{x_2}{y}\right)^2$ in accordance with DEHLAU's equation are plotted in the graphs of the test results (figs. 11 and 17). In most cases, it will be sufficient to obtain approximative values from the formula :

$$\left(\frac{x_2}{y}\right)^2 \approx \frac{1 + 4\, D^2_{ges}\, \lambda_{o_1}^2}{\left(1 - \lambda_{o_1}^2\right)^2 + 4\, D^2_{ges}\, \lambda_{o_1}^2}.$$

where :

$$D_{ges} = \frac{\rho_g}{2\,(\eta_1 m_1 + m_2)\, \nu_{o_2}},$$

and

$$\rho_g = \rho_1\, \eta_1^2 + \rho_2\, \eta_2^2.$$

η_1 and η_2 represent the proportions of the axle and bolster springing in the total springing; their sum amounts to :

$$\eta_1 + \eta_2 = 1.$$

A comparison with the test results showes satisfactory agreement in the case of tests 1, 2 and 3. The values found with the aid of DEHALU's equation are in even better agreement with the results of the tests, but the calculation takes more time.

Interpretation of the test results.

In figures 8 to 18, the results of the tests with the different springing arrangements are plotted graphically. In the curves marked 1, the movement of the car body is plotted at the ratio $\frac{x}{a}$, where x is the travel of the car body and a the amplitude of excitation, 5.65 mm. If the two axles were excited in the same phase, the curves would begin at the value

$\frac{x}{a} = 1$. But as the amplitudes of the two test stand axles are given a phase displacement of 90° in order to imitate passing over track irregularities, and as the excitation of the car body takes place through the bogie pin, the curves begin at the enlargement value obtained through dividing by $\sqrt{2}$, i.e. at $1/\sqrt{2}$ = approx. 0.7. The value for the amplitude of excitation is found from the relation :

$$a = \frac{a_0 \sin. \omega t + a_0 \sin. (90 + \omega t)}{2},$$

$$a = \frac{a_0}{2} (\sin. \omega t + \cos. \omega t),$$

where a_0 is the pre-set maximum amplitude of excitation at one axle. By substituting $\pi/4$ for ωt, the maximum value of a is found to be :

$$a = 0.7 \, a_0$$

In actual operation, the car body is of course not excited with such perfect periodicity and regularity as on the roller stand. Vertical vibrations in actual operation may be taken to be caused by the passing of the vehicle over a track irregularity which, in the case of railways, may be assumed to have the shape of a " sinusoidal curve " in the longitudinal profile, giving rise to the fundamental vibration wave (fig. 7). This notion corresponds closely to the phenomena at a rail joint when the rail is pressed down by the vehicle. The following equation applies to it :

$$y = -\frac{H}{2} [1 - \cos. (\omega t)].$$

where $\omega = \frac{2 \pi v}{L}$, with v = speed in metres pr.second, L = length of irregularity, H = height of irregularity, t = time in seconds.

If the irregularities are directly related to the length of the rail, the excitation is periodic, and L is equal to the length of the rail. Moreover, the fundamental wave is accompanied by an harmonic, approximately expressed by the equation :

$$y_1 = h_1 \cos. (2 \omega t).$$

One thus obtains, for the total profile curve :

$$y_0 = y + y_1$$

$$= -\frac{H}{2} [1 - \cos. (\omega t)] + h_1 \cos. (2 \omega t].$$

h_1 depends on the condition of the permanent way and is likely to vary between 0.1 and 0.5 H (mean value approx. 0.25 H). Where the permanent way is in very poor condition, h_1 may have to be assumed to be approx. 0.5 H.

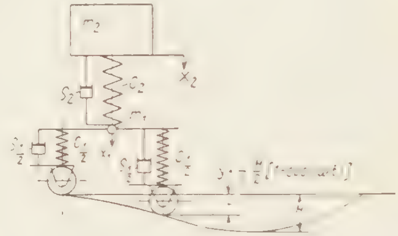


Fig. 7. — Profile curve of sinusoidal shape, generating the fundamental wave.

If ω is worked out for different speeds, one obtains frequencies which may vary, according to the length of the irregularity, between 1.5 and 10 cycles per second. As a general rule, however, with a speed of 100 or 120 km.p.h. and a rail length of 30 m, the main frequencies of excitation will be 1.9 and 2.2 cycles pr. second, respectively. As far as the conclusions to be drawn from roller stand tests are concerned, it is therefore the behaviour of the vehicles in the frequency range from 1.5 to 3.0 cycles which should be particularly instructive. The vertical running quality of a vehicle will normally be satisfactory in operation if, during the roller stand test with a frequency of about 2.0 cycles, the enlargement $\frac{x}{a}$ does not amount to more than about 0.5.

The curves marked 2 and 3 in figures 8 to 18 represent the movements of the axle and bolster springs, shown at the ratio

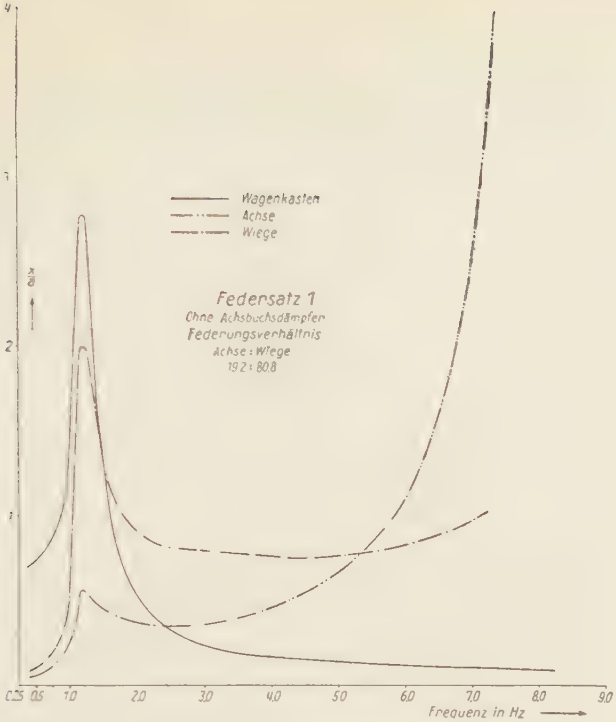


Fig. 8. — *Spring set 1*, without axle box damper; ratio of axle and bolster springing : 19.2 : 80.8.

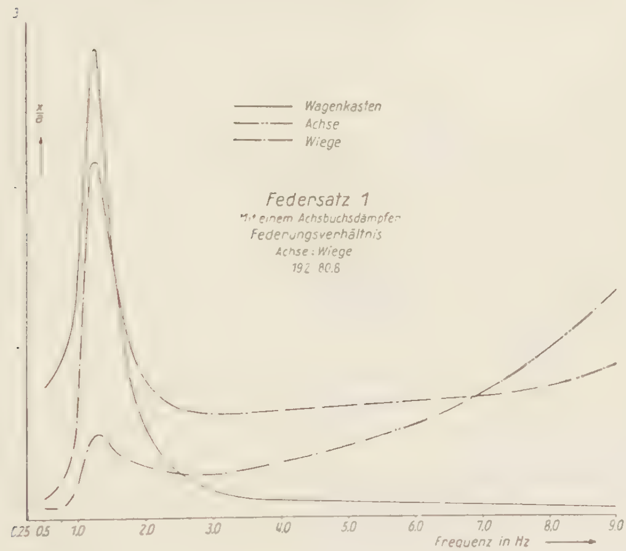


Fig. 9. — *Spring set 1*, with one axle box damper; ratio of axle and bolster springing : 19.2 : 80.8.

Fig. 8 to 18. — Results of vibration tests with spring sets 1 to 5 and different numbers of axle box dampers.

Explanations of German terms (fig. 8 to 18) :

Frequenz in Hz = frequency in cycles pr.second. — Federsatz = spring set. — ohne (Achsbuchs) dämpfer = without (axle box) damper. — mit einem (Achsbuchs) dämpfer = with one (axle box) damper. — mit zwei (Achsbuchs) dämpfern = with two (axle box) dampers. — Federungsverhältnis Achse : Wiege = springing ratio axle : bolster. — Wagenkasten = car body. — Achse = axle. — Wiege = bolster. — Gerechnete Resonanzkurve nach Dehalu = resonance curve calculated from Dehalu's formula. — Gerechnete Resonanzkurve nach Näherungsformel = resonance curve calculated from approximation formula.

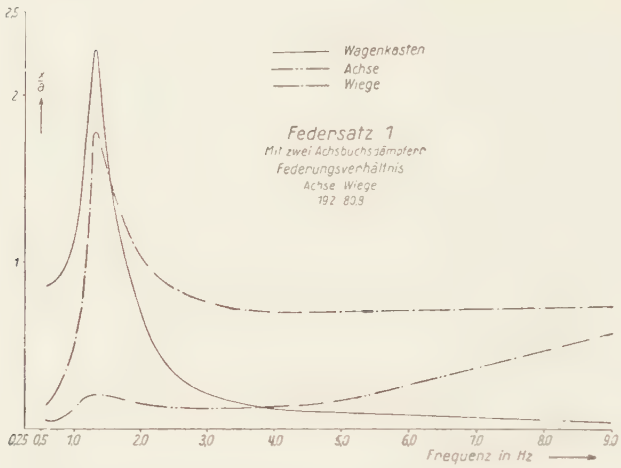


Fig. 10. — *Spring set 1*, with two axle box dampers; ratio of axle and bolster springing : 19.2 : 80.8.

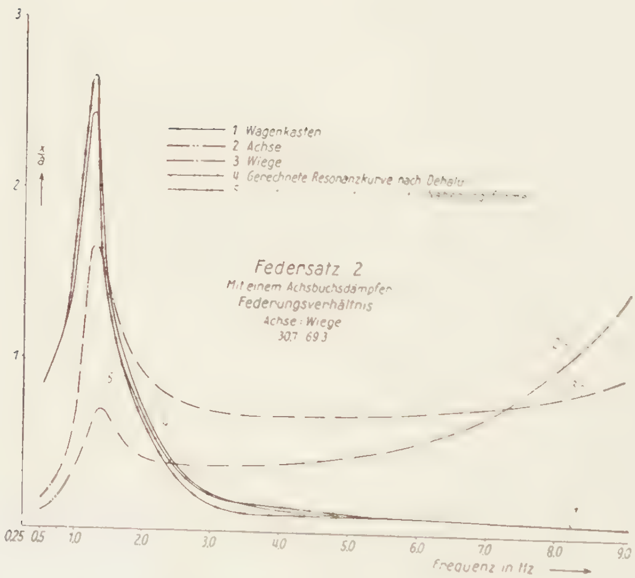


Fig. 11. — *Spring set 2*, with one axle box damper; ratio of axle and bolster springing : 30.7 : 69.3.

Fig. 8 to 18. — Results of vibration tests with spring sets 1 to 5 and different numbers of axle box dampers.

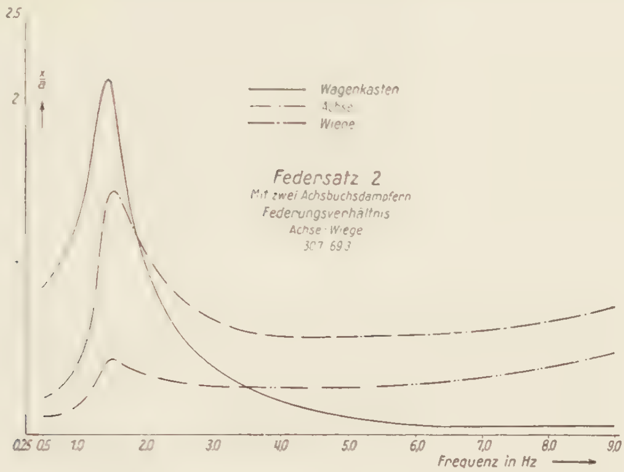


Fig. 12. — *Spring set 2,* with two axle box dampers; ratio of axle and bolster springing : 30.7 : 69.3.

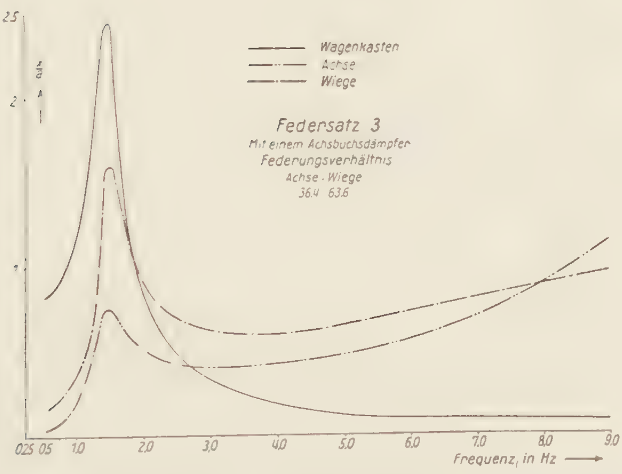


Fig. 13. — *Spring set 3,* with one axle box damper; ratio of axle and bolster springing : 36.4 : 63.6.

Fig. 8 to 18. — Results of vibration tests with spring sets 1 to 5 and different numbers of axle box dampers.

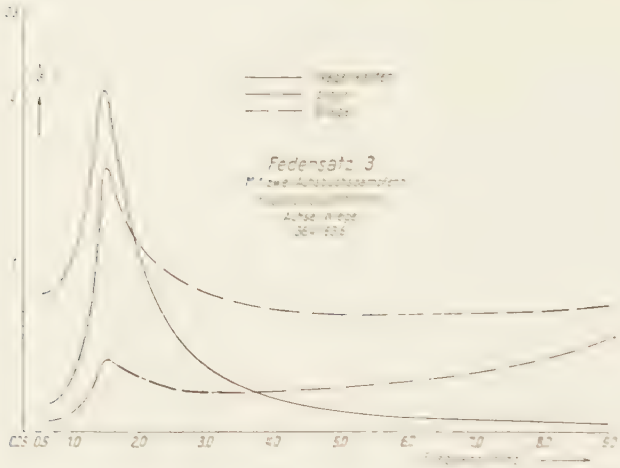


Fig. 14. — *Spring set 3*, with two axle box dampers; ratio of axle and bolster springing : 36.4 : 63.6.

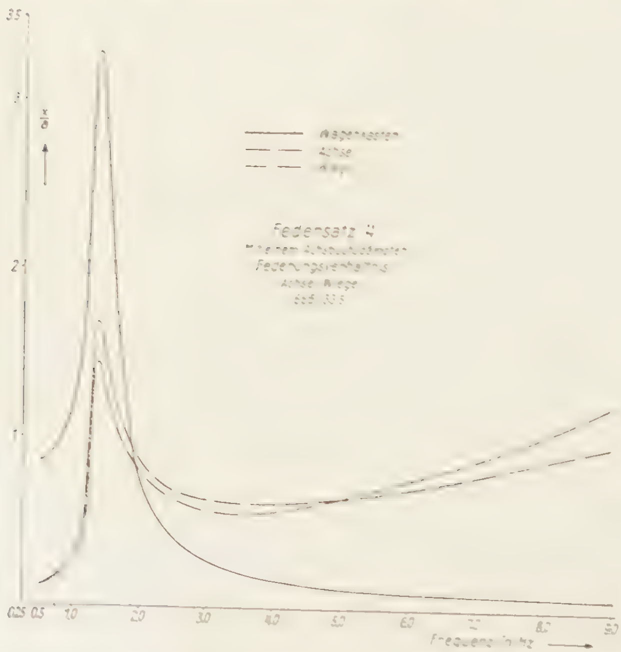


Fig. 15. — *Spring set 4*, with one axle box damper; ratio of axle and bolster springing : 66.5 : 33.5.

Fig. 8 to 18. — Results of vibration tests with spring sets 1 to 5 and different numbers of axle box dampers.

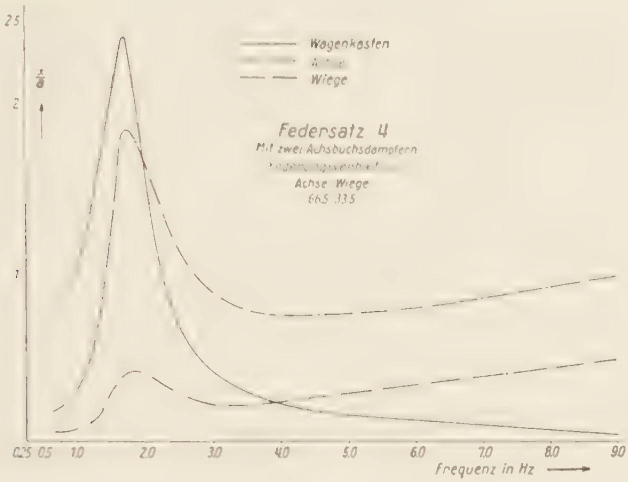


Fig. 16. — *Spring set 4*, with two axle box dampers; ratio of axle and bolster springing : 66.5 : 33.5.

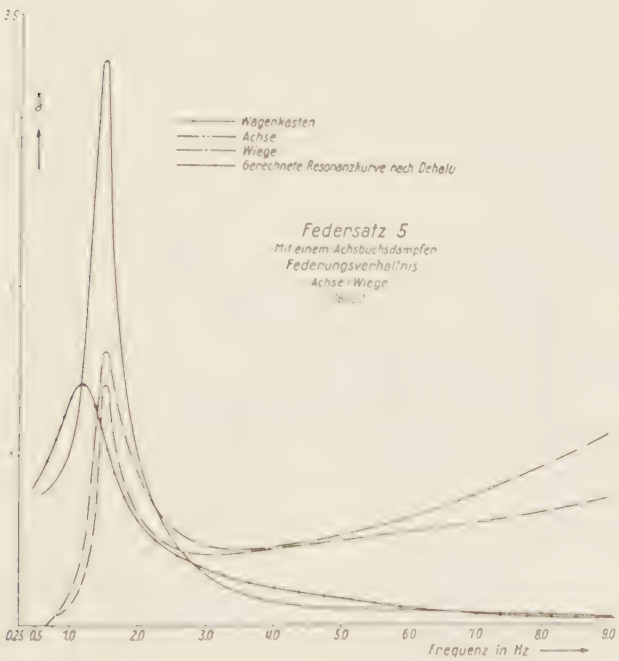


Fig. 17. — *Spring set 5*, with one axle box damper; ratio of axle and bolster springing : 78 : 22.

Fig. 8 to 18. — Results of vibration tests with spring sets 1 to 5 and different numbers of axle box dampers.

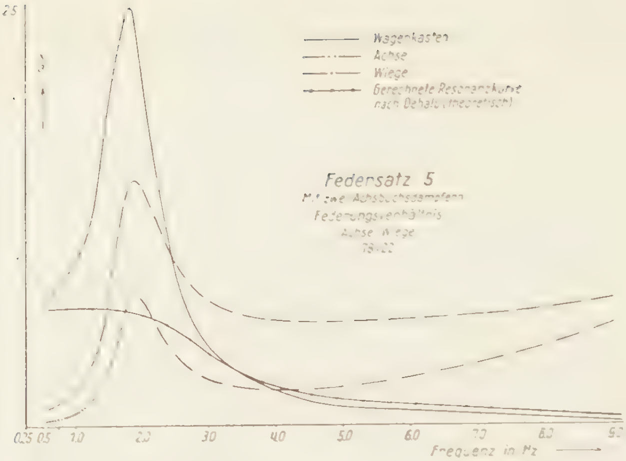


Fig. 18. — *Spring set 5*, with two axle box dampers; ratio of axle and bolster springing : 78 : 22.

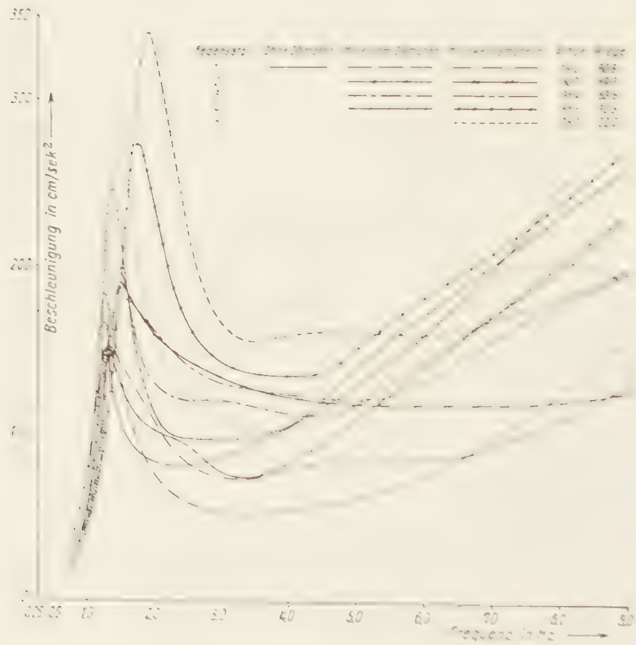


Fig. 19. — Acceleration, obtained from figures 8 to 18, for different spring sets and damper arrangements, assuming an amplitude $a_0 = 1$ cm.

Abscissae : frequencies in cycles pr. second.

Ordinates : accelerations in centimetres pr. second pr. second.

N. B. — Federsatz = spring set. — Ohne dämpfer = without dämper. — Mit einem dämpfer = with one damper. — Mit zwei dampfern = with two dampers. — Achse = axle. —Wiege = bolster.

$\frac{x}{a}$. The spring travel of car body and axle box springs should approximately correspond to the ratio of the spring constants. This is in fact the case in the lower speed range of curves 2 and 3 in figures 8, 9, 11, 12 and 13. But, as will be seen from figures 14 to 18, the resonance peaks move towards the higher frequencies if the spring travels do not correspond to the spring constants. This is due to the fact that, because the dampers ρ_1 and ρ_2 are (in the case of these bogies) arranged in series, the spring travels are no longer proportionate to the spring constant inasmuch as the stronger damper reduces the travel of the corresponding spring. The inherent frequency encountered in practice then depends on the actual spring travel.

If the ratios of the curves marked 1 in figures 8 to 18 are multiplied by ω^2, one obtains the ratios for the accelerations and, by assigning to them a certain amplitude, also the acceleration of the car body. Figure 19 shows the accelerations in the individual cases if the amplitude a_0 is assumed to be 1 cm. The illustration shows particularly clearly how the resonance peaks are moved in the direction of the higher natural frequencies. According to the curves, the most favourable spring ratio is that of case 1 (ratio of axle and bolster springing = 19.2 : 80.8) with one and with two axle dampers. With one axle damper, the resonance peak is admittedly somewhat higher than with two axle dampers. On the other hand, the curve for one axle damper has remarkably low values in the range of frequencies most commonly encountered.

Curve 4 in fig. 11 shows the motion of the car body calculated from Dehalu's formula, whilst curve 5 was calculated from the approximation formula developed above. It will be seen that the agreement of the theoretical values with the test results is satisfactory.

Curve 4 in figure 17 similarly represents the car body motion as calculated from Dehalu's formula. In this case, however, there is no longer agreement between the calculated values and those found in the tests. The discrepancy must be attributed to the shifting of the resonance peak due to the impeded springing. The axle dampers prevent the axle springs from acting in accordance with the spring constant. But, as a means of avoiding rocking vibrations, the axle dampers cannot be dispensed with.

Figure 18, too shows a resonance curve calculated from Dehalu's formula. Theoretically, this represents a very favourable curve, as there is no longer any resonance peak, and as the alignment of the curve between 1.5 and 3.0 cycles is also very favourable. In practice, however, this ideal curve cannot be obtained with the given type of bogie, for the reasons already stated.

Summary.

As a result of the roller stand tests with a Minden-Deutz bogie with different ratios of axle and bolster springing, the most favourable ratio was found to be about 20 % axle springing to 80 % bolster springing. In this connection, the damping must be so chosen that the axle box springing has a comparatively high attenuation constant and the bolster springing a comparatively low attenuation constant. D_{ges} should not exceed 0.2. In particular, the dampers should not impede the softer springing. With the given design of the bogies, this can well be realised if the ratio of axle and bolster springing is less than 20 to more than 80 %. With a large share of the axle springing, it is hardly possible to obtain a springing which is not hindered by dampers since, with soft axle springing, it will always be necessary to resort to comparatively strong dampers in order to reduce rocking vibrations. With the ratio of under 20 to over 80 % thus determined, it is advisable to choose D_1 = approx. 0.5 and D_2 = approx. 0.2. This corresponds to ρ_1 = about 100 to 120 kg sec./cm, ρ_2 = about 40 kg sec./cm, and D_{ges} = approx. 0.16. Here in are D_1 the damping factor, ρ_1 the damping resistance of the axle springing, D_2 and ρ_2 the corresponding values for the bolster springing, and D_{ges} the damping of the entire springing system.

[625 .1 (43)]

Raising the express-parcels passageway at Darmstadt Central,

by Dipl.-Eng. Alfred OHLEMUTZ, Frankfurt/Main.

(*Eisenbahntechnische Rundschau*, No. 9, September 1956.)

Changing to electric traction requires many alterations on structures to obtain the larger clearance necessary for the overhead wires. The raising of a large reinforced concrete structure described in this article is considered of general interest.

This re-inforced concrete covered passage-way was erected in 1911/12 over the lines in Darmstadt Central Station (fig. 1); it was divided by a partition along the longitudinal centre and was used both to give private access to the platforms for the Hessian Royal Family and their guests and by means of luggage lifts to ensure a quick route between the service platforms and the luggage office located in the higher level passenger station buildings. After 1918, the long vestibule of the former " Prince's Walk " was divided into offices and waiting rooms.

Fig. 1.

The building, which in effect comprises a single upper storey on reinforced concrete pillars, is about 112 m total length and is divided by expansion joints into four unequal sections : 17.35, 38.65, 36 and 20 m long. The frame is made up of two

outer single or double span beams, forming stressed walls of single storey height and having a small number of openings (windows and lift-doors). Between the main beams are the cross beams and intermediate longitudinals of the decking in the form of paving beams. The inner longitudinal partition, the ceiling under the wooden roof framing, the landings and staircases in the towers architecturally characterised by wide pediments on the south face (towards Heidelberg) are also of reinforced concrete. The part adjoining the station building was badly damaged in 1944 by air attack.

The problem.

With the electrification of the Heidelberg-Darmstadt-Frankfurt line, the free height under the bridge had to be increased by 0.60 m. It was not possible to consider lowering the track as the platforms were already 0.76 m in height. It was therefore first considered whether the entire construction could be demolished, and replaced by a steel bridge for baggage and express parcel traffic, 5.5 m above the tracks and the provision of other offices and staff amenities. Comparison showed, however, that the cost would be much greater than raising the existing structure, the 17.35 m section next to the station building, which had been damaged during the war, alone being demolished and completely rebuilt; this new section with a gradually sloping deck would then provide an approach ramp to the raised passageway from the despatch offices in the station buildings.

The baggage lifts on the northern face (Frankfurt direction) have not been retained; they have been entirely replaced as it was desirable to increase the capacity and weight; the lift motors have now been located at the heads of the shafts instead of in the passageway roof space.

The only document relative to the stability of the bridge which it was possible to find was a 1 : 100 scale drawing showing the diameter and number of main reinforcing rods. The decision to lift each section between the expansion joints separately was therefore more easily arrived at, as this drawing showed that the structure itself was sufficiently rigid to carry all the probable conditions which would arise during lifting. The very lightly reinforced pillars could not be considered as buried in the foundations, nor in the main beams, to form a portal. Calculations showed, however, that the bending stresses due to wind were fully compensated by the high compression stress resulting from its own weight so that there was no tensile stress in the pillar sections. To allow for lifting it was necessary to section 38 pillars. For this purpose, the concrete had to be broken at the line of the cut, the reinforcing rods severed and freed above and below by an amount equal to their diameter to provide for recovering the joint. After the post had been heightened by 0.60 m it was necessary to reinforce the intermediate portion with joint rods of the same diameter and also with rectangular bands and fill with B 225 concrete, taking particular care with the quality of the top joint and vibrating the casing continuously by Bosch hammers during the period of concreting.

Execution of the work.

After careful consideration of information supplied by tenderers regarding the methods proposed, the contract was awarded to "Grün & Belfinger A.G.". Under their proposal, the lifting would be done by ordinary hydraulic jacks, the structure before each lift being carried momentarily during the displacement of the jacks on prefabricated reinforced concrete blocks, 10 cm thick and the width of the pillar core slid into the space left between the pillar stubs. After insertion of the reinforcing extensions, these blocks were encased in the concrete. The Frankfurt Regional Railway Management as sponsors of the work specified the use of "perpetuum" jacks which did not involve displacement. This precaution was shown to be wise as it was found during the sectioning of the pillars that the rods were well embedded in an ample depth of facing concrete equivalent to B 225 quality, but the core concrete was hardly up to B 120, so that it would

have been impossible to allow its use as a solid concrete core whilst the reinforcing rods were re-covered.

The work started in May 1956. The first section, 20 m long, at the West end, was raised by the 27th May 1956, without difficulty. A month later the central section, 36 m long and 920 t in weight was done. The third and last section was set to its new height on the 29th July 1956.

During the construction work the luggage and express parcels service was served by a provisional luggage lift installed on No. 1 platform along the station building and by the barrow track at the South end of the platforms. The extension of the awnings of the longitudinal platforms beyond the high-level cross platform at the South (Heidelberg) end was terminated at the parcels bridge. The purlins and deck beams are supported by the North wall of the bridge. Temporary showing with wood scaffolding presented no difficulty, but was relatively costly.

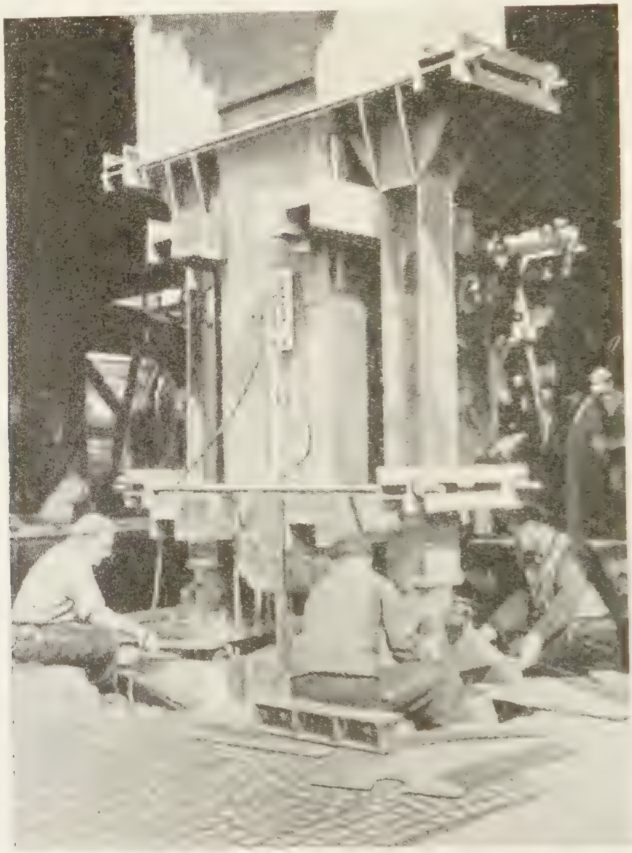


Fig. 2

The main beams of the bridge, which rested on the pillars have gussets or bottom ribs of arch shape. The auxiliary pillars had to be set in line with these gussets or arch capping next to the main pillars. To

transfer the pressure it was thus necessary to fix solid wedges of reinforced concrete to the other concrete by means of anchors and reverse notching to avoid shear, as shown in figure 2.

The IP or steel posts were solidly anchored against tension and compression with the portion of the post to be raised. In most cases one auxiliary post was sufficient, only in the case of a few heavily loaded pillars was it necessary to use them on both sides.

The double posts were carried on the jacks by arrangements similar to sub-beams. The foundations for the jacks and auxiliary pillars were concreted on the existing pillar foundations with shear anchors, and lighting reinforced. It was necessary here to accept light stresses eccentric to the foundations because of the efforts caused by the pillars. As the width of the gap for lifting was a uniform 612 mm at the end of the job and corresponded to the amount of lift relative to rail level, it could be accepted that there was neither subsistence nor movement of any foundation.

For the lifting operations, which were done on Sundays (when there was no business traffic), all the auxiliary pillars were put into position and the "perpetuum" jacks placed under the pillars. Each of the jacks was connected to a separate oil pump; from the contractor's experience this method is better than a single oil supply from one pump driven by a motor with separate pressure regulation for each jack and avoids the need for long feed pipes.

The day before the lift all the jacks were put under pressure and alternate pillars sectioned, the others being notched sufficiently to allow them to be completely severed within half-an-hour, including the curring of the reinforcing rods, immediately before raising.

Control of the operation.

Because of the separation of the pumps it was necessary to arrange for the lifting operation to be done uniformly, in view of the stroke length of the " perpetuum " jacks, which is about 120 mm, it was necessary to avoid overloading or relieving of some auxiliary pillars which would have caused undesirable displacement of stress in the structure. The limit of irregularity had been fixed at 5 mm; even though it was sometimes unavoidably exceeded there were neither critical nor any other points, any signs of cracking, breaking away, peeling or similar conditions.

The uniformity of lifting was controlled with sufficient accuracy by means of a single electrical indicator designed by " Grün & Bilfinger " : on each pole were fixed five small indicator lamps of different colours, on a board; amongst them was a very bright " stop " light. The small lamps were connected to points on a contactor board, with 6 mm gaps. When the lifting brake was opened out another small contact with a roller fixed to the foundation slid over these contacts in such a way that, allowing for the width of the contacts, the following lamp was lighted at the end of a bore 5 mm lift. In addition, a movable reglet scale was fixed and set before a lift so that it was possible at any time to see the total lift at each pillar. The pump operators and the supervisors of a group of pumps for 3 or 4 pillars could thus constantly verify the lifting operation for themselves. However, it was essential to verify from a single point that no jack was lifting too quickly or too slowly. The indicator lamp panels of all the pillars were therefore duplicated on a large control table (fig. 3). From this point the controller could signal the operator of a pump or the supervisor of a group to stop if they were in advance or behind the other jacks. A master switch allowed all pumps to be stopped simultaneously. Resumption of lifting was signalled by electric bell. For communication with certain work positions loud-speakers were installed and these were perfectly satisfactory. Generally, the controller had only to watch that the lights were as nearly as possible simultaneous in operation and give anyone in advance, which could easily be seen, an early signal to stop.

The average time for a full lift of 6 mm was about one minute, the interval between the first light and the last in a series was

hardly more than 40 sec; in view of the inertia and rigidity of the reinforced concrete, this difference caused no anxiety.

When the lift had reached a total of 132 mm red lights were illuminated to show that is was necessary to give the jacks an intermediate support as they were near to release the piston and return it to the starting position by pumping the oil from the feed pipe. Whilst the supply lines were relatively short, this operation took a certain amount of time particularly with the old type jacks which had four outside helicoidal springs. When the piston was fully retracted, the withdrawal which now amounted to

Fig. 3.

the limit of their stroke. With the " perpetuum " jack, of course, there is no need to remove the machine as the cylinder of the lifting is designed so that it also forms a footing for releasing and withdrawing the piston. For safety, the base of the jack, which was driven during the lift, was progressively wedged by plates of thicknesses varying from 2 to 20 mm (fig. 2) so that in the event of sudden collapse of a jack, the drop would not have been more than a few millimeters. When a stroke of 132 mm had been reached, the intermediate plates were replaced by two prepared IP 12 blocks which were slipped under the base of the jack to form a new bed; it was then necessary upwards of 120 mm was completed by slipping three more IP 12 blocks under the cylinder on which the piston could then bed for the following part-stroke to be made. A clearance of 12 mm was necessary for this purpose. This thus gave a total movement of $5 \times 120 + 12 = 612$ mm. When the lifting was completed, the jacks were on five crossed layers of IP 12 blocks, bolted together on two plates of 10 and 2 mm respectively. The resilient compression of the sub-foundation was insignificant.

Naturally, the lifting was also checked by level. At important points of the new structure gauge, above the track prior to raising bench maıks were placed to fix the

datum level and to ascertain its relative height above the reference level before the lifting operation : it was thus possible immediately after lifting and again later, after removal of the auxiliary pillars, to confirm the height of the bridge.

The execution of one stage of lifting of 132 mm and the withdrawal of the ram and re-shoring took about half an hour; for the 612 mm lift, therefore, an uninterrupted period of five hours was needed. Each oil pump was operated by two men, the two double-ram pumps by four. For the continuous shoring of the jacks, two men were needed for each set of intermediate plates. In addition, there were five supervisors for the five groups of pillars separated by tracks. In total, sixty-five men were needed including overall supervision, recording, Maihak extensometer control and critical point support, pump maintenance, jack maintenance, signals installation, levelling and miscellaneous duties as required.

During the lifting operation, the stability of the part of the structure to be raised perpendicular to the longitudinal axis was maintained by means of a solid assembly of round bars under the structure. It was fixed at the lower part by oblique ties which were adjusted during the lift in accordance with the amount required. In addition, when inspecting a bench mark through the eyepiece of a level instrument, any possible transverse movement would be shown.

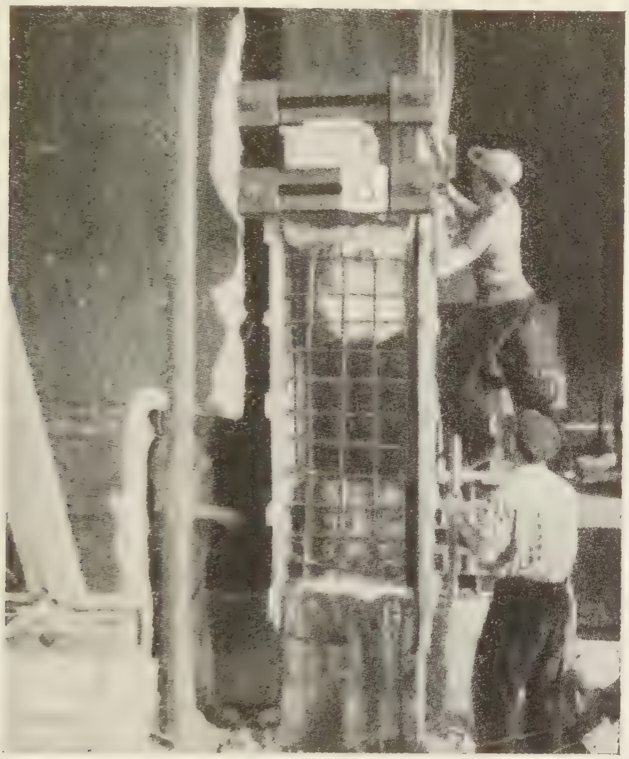


Fig. 4.

Final work.

After the lift, the structure was carried on the auxiliary pillars and on the cylinders of the jacks. The pistons of these were free, the pumps could be disconnected and removed. At this point, the remaining central parts of the pillars, if they were made of low strength concrete, were broken down and the bearing surface prepared to take the new cover, etc. The reconstruction of the pillars extended over about 2 m high in the most suspect portion; in addition to the iron bars which overlapped for the lengthening, an adequate banding strap (fig. 4) was also put in, thus remedying a deficiency in the original pillars.

The auxiliary pillars and the jacks were the same for the three sections of the lift. To be able to lift one section four weeks after the preceding one, it was necessary to be able to transfer the load of the bridge out to the newly concreted pillars.

No. 325 metallurgical cement was therefore used and transfer of the load commenced after not less than eight hours. Because of the rigid link between the bridge and the auxiliary foundation which, even to the jacks, was no longer able to be freed, the disengagement of the auxiliary pillars was started by breaking down the reinforced concrete wedges fitted to the gusset of the main beams, without observing any particular order of succession. This method did not give rise to any difficulties.

The reconstruction of the fourth section of the structure, along the station building, did not present any difficulty apart from those to be expected normally in the erection of a reinforced concrete bridge of 17.5 m span over railway tracks. It is therefore not necessary to spend any time over this part of the work.

The raising of the baggage and express parcels bridge at Darmstadt Central Station provided one of the most interesting tasks of the series of operations of the same kind made necessary by electrification work in the area covered by the Frankfurt Region of the German Federal Railways. The traveller, entering Darmstadt Station from the direction of Heidelberg, will hardly notice that it has been necessary to raise a complete building, crossing ten tracks and five platforms, to allow the catenary to pass underneath it.

[625 .212 (493)]

Three years experience of grinding tyres on the Belgian National Light Railways (S. N. C. V.),

by Robert MORNARD,

Ingénieur principal, Chef de Service à la Société Nationale des Chemins de fer Vicinaux.

(*Revue de l'Union Internationale des Transports Publics*, Vol. V. No. 3, December 1956.)

I. INTRODUCTION.

Some years ago when considering the reorganisation of repair shops, the technical services of the S. N. C. V. found that tyre wear was a predominant reason for sending railway vehicles to shops for lifting. In other words, the tyre was the « low spot » to returning the vehicle to shops.

For this reason, they fastened their attention on this problem and found that maintaining the tread of the tyre in perfect condition would bring them many advantages.

a) Better contact between the wheel and the rail.

The tyre, the part of the wheel in contact with the rail, theoretically is in the form of a circle.

The circumference, the tread, situated in a plane rigorously perpendicular to the axis of the axle, does not retain this form for any length of time and shows positive or negative departures from the initial design.

In this connection, it is interesting to record that the tyres in the P. C. C. motor coaches not fitted with brake shoes show a greater degree of ovalisation than the others.

This ovalisation is aggravated by the formation of a number of short flats resulting from starting and braking and also from the variation in die quality of the material of which the tyres are made.

With a slight exaggeration it can be said that the initial circumference in the long transforms itself into an elliptical polygon.

Furthermore, in spite of the care taken in selecting tyres of closely similar hardness for each pair of wheels, the treads after running some ten thousand kilometres differed in diameter and the difference became more marked as the mileage built up.

All the above phenomena set up at peak speeds serious vibration.

This resulted in dynamic fatigues acting on the track and details of the rolling stock (brake gear, brushes, traction motors current pick up, bearings, brasses, etc.).

In addition, we think we have to see here a factor causing the phenomenon of corrugated rail wear.

The S. N. V. C. technical services consulted the German railways, who for a long time have used the double method of grinding tyres and rails.

This latter operation is absolutely indespensible as there takes place a « wheel-rail » action of reversible harmfulness.

There is no disputing that the improvement of this wheel-rail contact has given our passengers greater comfort by noticeably sweeter running.

b) Reduction in the cost of track and rolling stock repairs.

Three years use of this practice in the Hainaut group of lines has brought us great financial satisfaction in this field.

c) Increase in mileage run between two lifts.

As we said, tyre wear was the « low point » on which vehicles are reshopped.

Formerly, the tyres were turned up after running 40 000 to 90 000 km (24 850 to 55 920 miles) according to the type of vehicle, the topography of the localities and the urban or suburban character of the lines.

When the vehicle went into shops after 40 000 km it was given a light repair, but after 80 000 km (49 710 miles) a heavy repair.

To what must these abnormal wears of tyre treads be attributed?

1. Axles not being parallel one to the other.

2. Bent axles.

3. Form of the normal tyre profile.

4. Badly maintained track and curves without transitions.

5. Deformed circle of tread.

6. Unequal diameter of the tyres on the same pair of wheels.

7. Difference in hardness of the two tyres of the same wheelset.

8. Form and hardness of the brake blocks, etc.

These wears are characterised relatively to the initial profile by a thinning of the flange and by a flow of metal giving a « false flange ».

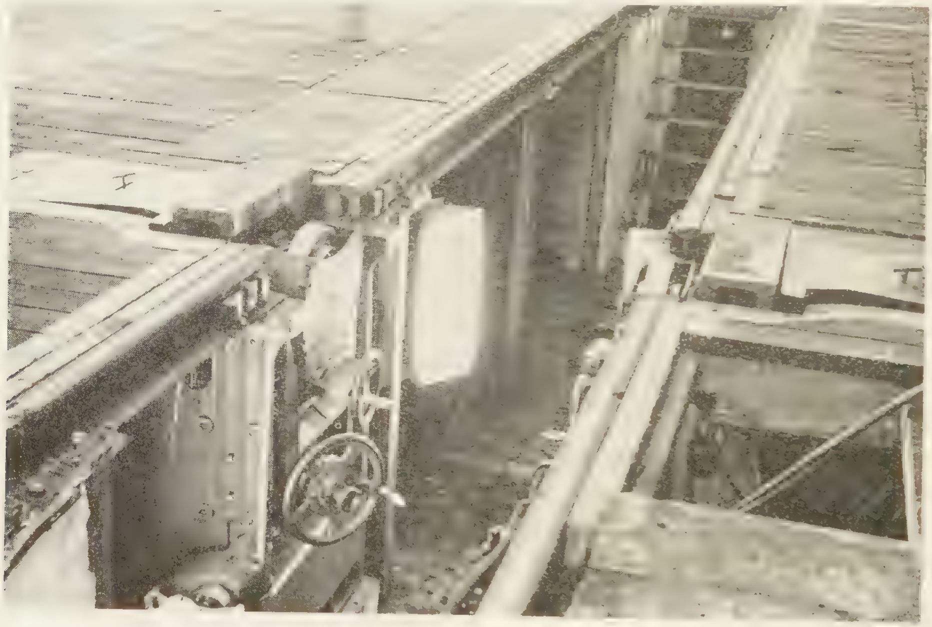


Fig. 1. — Machine for grinding tyres.

From this can be deduced that the effects of the causes which degrade the tread ought to be countered or at least reduced periodically. This is what led our Company to introduce the method of grinding tyres.

The first installation was put down at Charleroi in this Hainaut group, an important electric traction holding of the S. N. C. V.

We soon found that an important number of the flanges remained almost intact as the mileage piled up when the treads were periodically corrected gy grinding.

The remainder had the flanges built up and these alone were returned.

Economically speaking, this practice as we shall see later will give large savings.

Over a year ago, our Antwerp and Brussels groups acquired similar equipment.

We expect during the next financial year to put down a grinding plant for our Belgian coast system. We think that in this last case although the mileage is relatively reduced, the equipment will pay for itself and the first cost will be amortized in five years; actually the sand makes the wear of tyres particularly acute.

If we have succeeded in raising this « low spot — tyres », we have to face other problems; these are mentioned briefly in the conclusions.

* * *

We give a view herein of the grinding plant at Charleroi.

Two grinding wheels are visible at the moveable lengths of rails; to make the photograph clearer, the two supports which raise the body have been put in the upper position (Photo : « Nos Vicinaux »).

* * *

II. COMPARATIVE GRAPHICAL STUDY OF THE LIFE OF A TYRE REPROFILED OR GROUND.

The following figures and formulae relate to a definite type of rail motor coach (Standard metal bogie).

We will explain the method used so that the reader can readily include data from his own working.

Let use consider the profile of a tyre :

Let E be the useful thickness of the tyre (in mm) that we can utilise without exceeding the safety limits.

For the type of stock selected :

diameter of tyre (new-profiled) . 680 mm
scrap diameter . . . . 600 mm

From this is deduced E = 40 mm.

What are the factors tending to reduce the value of E ?

a) We find that immediately after reprofiling whether it be an unmachined tyre or a tyre having already been in service, there is a closing together of the metal to the extent of a reduction of 1 mm on the diameter after running a few hundred kilometres; this phenomenon is known under the name of hammer hardening of the tread.

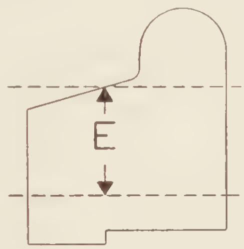


Fig. 2. — Cross section of tyre.

We deduce that through the consolidation of the tread E has been reduced by 0.5 mm.

Though this phenomenon may not be instantaneous, we will consider it as being so in diagram I (fig. 3).

N. B. — We can now make an important remark on this subject : at each reprofiling in the life of the tyre, we experience this phenomenon of draw down whereas by grinding when only a thin layer of the order of 0.3 mm is removed, it does not occur.

b) Experience shows that the wear of the tread follows an exponential law and not a linear one.

Let us try to arrive at it.

We have wear = U (function of T : number of kilometres in mm.

$$U = 0.5 + A.T.^{B}$$

T expressed in 10^3 km.

The statistics give for :

T = 20 000 km U = 2.5 mm
70 000 km 10.8 mm

There are thefore two equations with two unknows A and B.

$$A.\ (20)^{B} = 2 \qquad (1)$$
$$A.\ (70)^{B} = 10.3 \qquad (2)$$

Dividing (2) by (1), we get :

$$(3.5)^B = 5.15$$

or $B = 1.308$ and $A = 0.04$.

From this we conclude that the curve of wear after T thousands of km will be of the form :

$$\boxed{U_{(T)} = 0.5 + 0.04\ T^{1.308}}$$

This curve has been traced in diagram I.

c) We can, for every period T, determine the mean value of the number of km required to wear away 1 mm of tyre.

Let us call « *a* » (in thousands of km per mm) this average value.

We have :

$$\boxed{a = \frac{T}{U} = \frac{T}{0.5 + 0.04\ T^{1.308}} = \frac{1}{\frac{0.5}{T} + 0.04\ T^{0.308}}}$$

Let us find the value of T for which « *a* » is maximum; let $da/dT = 0$.

We find T = *17 000 km.*

The curve « *a* » as function of T is plotted on diagram I.

d) Returned tyre.

Let us call :

E (in mm) the useful thickness of the tyre to be used;

N the number of returnings during the life of the tyre;

e (in mm) mean thickness removed when returning;

T the number of thousands of km at which the tyre will be returned;

x the number of km measuring the life of the tyre.

Formulae : (Do not forget that after each returning there is a shrinkage of 0.5 mm on the radius).

$$\begin{cases} N.e + (N + 1)\ (0.5 + 0.04\ T^{1.308}) = E & (1) \\ x = (N + 1)\ T & (2) \end{cases}$$

N should be a whole number.

To give a numerical example :

E = 40 mm $\qquad$ *e* = 7 mm

Take as an approximate value of T = 70 000 km.

These values are used in (1) and we calculate N (taking the next lower full number).

We find N = 1.

Using N = 1 in (1), we solve the exponential equation, which gives the definitive value of T.

We find T = 98 000 km.

From this we deduce that the mean life of the tyre will be 98 000 × 2 = *196 000 km.*

e) Ground tyre.

We will use the same notations as for the returned tyre but this time :

N will be the number of grinds;

e, the average thickness removed during a finished grind;

T the number of thousands of km at which the tyre will be ground up.

Important note. — Seeing that at each regrind only the hardened skin is removed, the value of 0.5 mm is only to be taken into account when a new tyre is being put into used.

Formulae :

$$\begin{cases} N.e + 0.5 + (N + 1)\ (0.04\ T^{1.308}) = E \\ x = (N + 1)\ T \\ N \text{ whole number.} \end{cases}$$

Numerical example :

E = 40 mm $\qquad$ *e* = 0.25 mm.

We proceed in the same way as before, taking T = 20 000 km.

We find N = 17, whence T = 19 500 km

rounded off to 20 000 km, whence $x = 18 \times 20\,000 =$ *360 000 km* approximately.

This is about 1.8 times the life of the tyre returned to profile.

To bring this home more forcibly, we have drawn diagram II by which we can compare the curve of the life of a tyre which is turned up and that of one reground.

After this last operation, carried out to restore the tyre to its original profile, the difference amounts to several millimetres.

III. FINANCIAL ASPECT OF THE PROBLEM.

Using the two examples we have given, we will now evaluate the cost price per km of the tyre reprofiled by machining and by grinding :

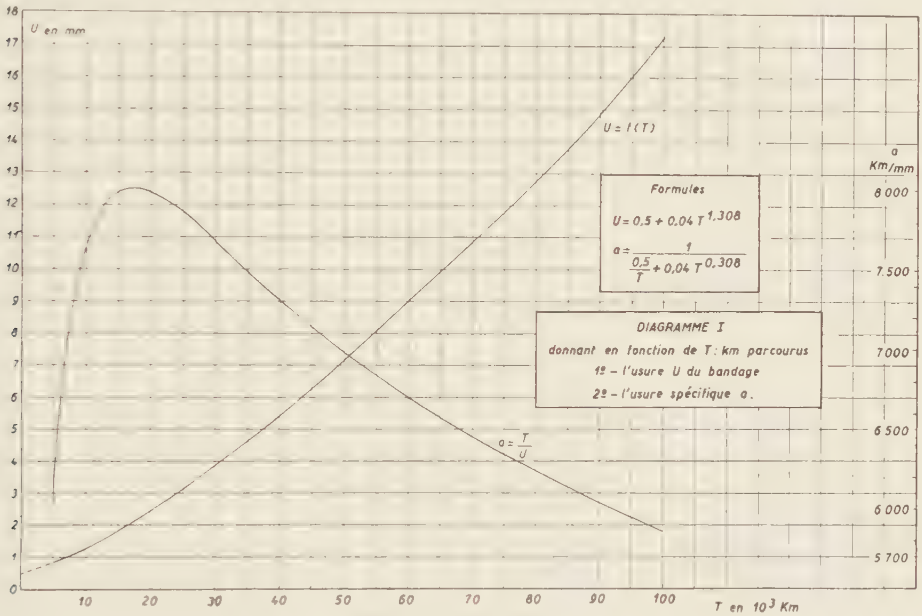


Fig. 3. — Diagram I.

N. B. — U en mm = u in mm. — Formules = formulae. — Diagramme I donnant... = diagram I giving in terms of T the km run. — 1) U the wear of the tyre; 2) A specific wear *a*.

This diagram deserves some comments :

1. The curve of material used or removed from the ground tyre is always lower than that of the tyre returned (except at the time of the first grinding when it is the same as the layer removed in turning).

2. At the time it is returned, the tyre which is not ground already shows greater wear than a ground one and *this before it is returned to profile.*

a) *Tyre restored to profile by turning*

Let us take the average price of the re-turned tyre (price of the new tyre, boring, turning, reprofiling, etc.) as 2 900 Belgian francs.

The price of the km-tyre will be

$$\frac{2\,900 \text{ fr}}{196\,000} = 0.0148 \text{ fr.}$$

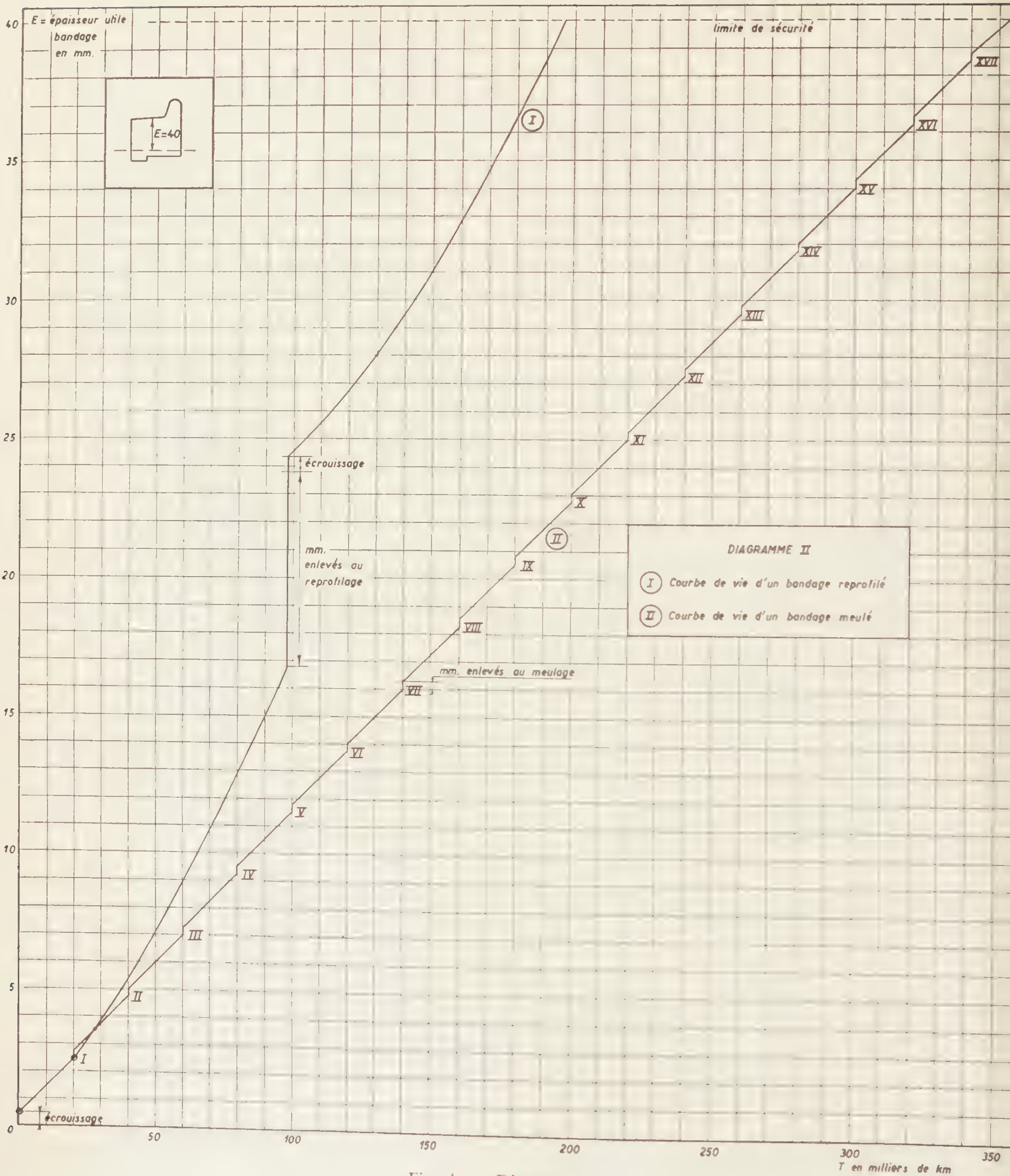


Fig. 4. — Diagram II.

Explanation of French terms :

Limite de sécurité = safety limit. — E = useful thickness of tyre in mm. — Ecrouissage : mm enlevés au reprofilage = mm removed by reprofiling. — Mm enleves au moulage = mm removed by grinding. — I = life curve of a reprofiled tyre; II = life curve of a grinded tyre.

b) *Ground tyre.*

Let us take the average price of the ground tyre as 3 260 Belgian francs. The price of the km-tyre will be

$$\frac{3\,260 \text{ fr}}{360\,000} = 0.00905 \text{ fr.}$$

that is to say a difference of 0.00575 fr. per wheel per km.

In the case of a motor coach, this saving is 0.046 fr. per coach per km.

The conclusion to be drawn from all this is that the savings realised by the « tyres » centre of an operating unit running 5 000 000 km (3 106 000 miles) annually would be about 230 000 Belgian francs.

To this saving, we must add that realised on labour costs due to the increased mileage between overhauls.

Then too, as we said above, should the flange show some wear, it was built up and this flange *only* would be restored to the correct profile in the lathe (without cutting into the tread).

The financial gain from this is evident; in fact if through this building up, we avoid having to remove 5 mm from the tread we get a saving of :

$$5 \times 60 \text{ fr.} = 300 \text{ fr. per tyre.}$$

Number of years to write off the capital cost of a grinding equipment.

At the time of acquiring such equipment, it is well to remember :

1. The purchase price of the machine which we will call . A francs.
2. The cost of the grinding pit (concrete foundation, auxiliary electrical equipment, lighting, forced suction gear) B francs.

   The sum (A + B) in Belgium today lies round about 700 000 Belgian francs (seven hundred thousands).

3. The value of the annual saving :

   on materials
   on labour } . . C francs.

   Let :

r : the rate of interest per year of the capital invested and of the annual saving (compound interest) (r is expressed in hundredth, per example : 0.05);

n : the number of years to be found

We have :

$$\boxed{C \frac{(1 + r)^n - 1}{r} = (A + B) \quad (1 + r)^n}$$

making $x = (1 + r)^n$

we get $$x = \frac{C}{C - r\,(A + B)}$$

Knowing x all that is needed is to solve the exponential equation $x = (1 + r)^n$ to find n.

Example :

$$\begin{aligned} A + B &= 700\,000 \text{ fr.} \\ C &= 260\,000 \text{ fr.} \\ r &= 0.05 \ (5\ \%) \end{aligned}$$

from which :

$$x = \frac{260\,000}{260\,000 - 0.05\,(700\,000)} = \frac{260\,000}{225\,000} = 1.155$$

A compound interest table gives us $n = 3$ years.

IV. CONCLUSIONS.

We have mentioned the great satisfaction we have experienced from grinding tyres both from the financial and the technical aspects.

We have not touched upon the technique properly speaking of the process as it is sufficiently well known. Nonetheless, the starting up of such a plant is not done without difficulties which have to be overcome.

We have found that the organisation of

a large number of rail vehicles (200 bogie motor coaches and 100 bogie trailers) often dispersed over a radius of 50 km (31 miles) needed strict discipline sometimes difficult to maintain, and more so in the period of winter damage.

And yet experience demonstrates that if a motor coach exceeds by 10 000 km (6 200 miles) or even 15 000 km (9 300 miles) the period laid down for regrinding, it is difficult to restore the tyre.

But like the barrel of the Danaïdes, our technicians have to solve a number of problems.

They have lifted up the « low point » of the tyre to 120 000 km (74 500 miles); other parts which wear, such as carrying bearings, brake gear, etc., are now receiving their attention.

The question now raised is the following : must we retain a useful thickness of tyre imposed by the Purchasing Department and risk wasting some millimetres or should we calculate the thickness of tyre which would let the vehicle run a certain number of kilometres between lifts compatible with the other « low points » of other wearing parts, so as to reduce to the minimum unusable millemetres of tyres.

So many questions and so many answers which have to be dealt with and answered by a rational solution if the saving obtained by grinding the tyres is not to be lost.

[656 .1 (73)]

Extension of the organisation for hiring automobiles in the United States.

(*Internationales Archiv für Verkehrswesen*, No. 6, 2nd Part, March, 1957.)

Sometime ago, the American Press announced that the important firm in the hiring of automobiles, the « Hertz Corporation », had just ordered from various American Builders 15 600 new motor cars of a value of 33 million dollars.

In spite of the fact that at the present time one American out of three has his own vehicle, the hiring of automobiles has become of considerable importance in the United States. Besides the three large undertakings represented in all parts of the country, there are in addition about 100 firms hiring cars on a smaller scale. In all some 200 000 vehicles are running at the present time on hire service.

The most important firm engaged in hiring vehicles and in existence over 33 years is the « Hertz Corporation », which has more than 1 000 agencies spread over some 700 American and Foreign Agencies. Its stock is made up of about 16 000 motor cars of all types and classes. The second firm is the « A.V.I.S. Rent-a-Car System » with 10 000 cars for hire in 950 towns, whereas in the third rank is found the « National Car Rental System Inc. » with also 10 000 cars distributed over 350 places.

The price of hiring amounts at the present time on an average of 7 to 8 dollars plus 7 to 8 cents per mile run. Without the additional charge per mile the minimum price per week varies between 35 and 45 dollars. Included in the price are the operating cost, petrol and oil as well as insurance.

To hire a vehicle all that is required besides a good appearance is an identity card and a valid driving licence. Some firms require in addition a deposit or guarantee of 20 dollars for hiring by the day and 60 dollars when hiring by the week. Nonetheless, recently, there is increasing tendencies to give up this practice. The reason is partly that thefts of hired cars have become extremely rare and secondly that many customers possess a document known as a Credit Card delivered in the name of one or several hiring firms and which is considered practically as the client's card. Messrs. Hertz for example estimate that about 1.7 million of these cards in their name are in use, some of which were delivered 30 years ago. Moreover, the make up of the present clientele is an argument in favour of the general abandonment of the system of requiring a deposit or guarantee. Whereas at the introduction of the practice of hiring out vehicles most of the customers had the sole desire of finding themselves at the steering wheel of an automobile, today it is mainly for business purposes that lead to the conclusion of a hiring contract. Taking as a whole, 80 % of the customers are business people who are on a business tour and only 13 % hiring vehicles at the present time with the object of making an excursion. As regards the feminine clientele, this averages about 15 %.

The particular importance of the firms hiring automobiles in the United States is due to the fact that the service to the customers is to be found everywhere and has been based on a large scale. For example, the « A.V.I.S. System » organisation offers to their passengers the means of hiring a car for only 10 dollars per day, the vehicle being available at the aerodrome when the customer leaves the aircraft. The price of hiring covers journeys up to 50 miles.

The « Hertz Corporation » on the other hand has devoted its activities more particularly to passengers by rail. In about

300 stations in America and Canada, this firm has installed its own telephone kiosks from which the customer can speak directly to the firm's office and take possession of a vehicle within a few minutes of his arrival.

These hire services recently have been extended to countries outside America. The « Hertz Coporation » for example not only hires the vehicles in most foreign cities but undertakes to provide the customer with all the necessary documents such as an international licence and an insurance card when the client in any foreign locality takes possession of the motor car ordered.

In addition to these various inovations, there is also the service known as « city to city » by which the customer has the convenience of hiring the vehicle in one city and giving it up at his destination which gives him an appreciable saving on the hire charges in the event of him not having to make the return journey immediately. About one third of the customers only rent a vehicle for a day, another third for periods of 2/5 days, whereas the remaining third make contracts to hire for one week or more.

Most of the vehicles on hire are of the most recent types and are given up by the hiring firms after running 18 to 20 thousand miles.

The organisation of car hire is at the present time in the United States enjoying great prosperity.

It is interesting to know that three quarters of the number of vehicles hired in the United States are delivered to airports for the customer hiring them.

NEW BOOKS AND PUBLICATIONS.

[**656** .2 (4)]

Vereintes Europa auf der Schiene (*Europe United by the Rail*) Special edition of the Periodical (*Europa-Verkehr*). Published under the Editorship of Prof. Dr. Ing. E.h. FROHNE, with the collaboration of many German and Foreign Authors. — 1957, May, Darmstadt, Otto Elsner Verlagsgesellschaft. Format DIN A 4, 186 pages with 67 illustrations and tables. Price half-bound DM 12. —; special price for subscribers to the paper *Europa-Verkehr ;* DM 8.50.

This special number of « Europa Verkehr » (European Transport) is intended to bring out the influence of railways in the unification of Europe.

In the preface, Dr. Eng. SEEBOHM, Minister of Transport of the German Federal Republic and President of the Confederation of European Transport Ministries, refers to the principal creations of railway undertakings in the international field. In their technical. commercial and legal characteristics, certain of them go back very far into the history of railways.

The publication collects together a large number of notes written by leading personalities in the transport world. These notes deal with all subjects relating to international transport relationships. Their object is not only to depict the present situation with the results obtained but further and above all to indicate in what directions a new effort should be tried to increase the rapidity, efficacy and economy of transport.

Two notes which can be said to be introductive are due to two of the Directors of U.I.C. (International Railway Union) whose intervention frequently and very decisively in the field covered will be referred to several times in the pages of this work. Dr. Ing. E.H.E. FROHNE, the first President of the German Federal Railways, Vice-President of U.I.C. brings out the points which differentiate the European Railways from one another and what ought to be done to obtain the desired unity. M. L. ARMAND, President of the Société Nationale des Chemins de fer Français, President of U.I.C., demonstrates the necessity for collaboration between the European Railways, and gives as points of support the work already done towards European integration.

On the technical side will be found investigations into the permanent installations : the superstructure of the track, the modern structures and signalling equipment.

Suggestive points of views are evoked in regard to the rolling stock : electric locomotives and rail motor coaches, Diesel traction, weight reduction of the vehicles, standardisation.

New ideas are brought forward as regards timetables for international trains both passenger and goods.

The better utilisation of the rolling stock is favoured by the setting up of the « *Europ* » stock of wagons and by the « *Interfrigo* » (International Railway Transport in refrigerated wagons). The financing of the standard type of vehicles will be arranged by the setting up of « *Eurofina* » (European Company for Financing Railway Rolling Stock). The containers have themselves passed across the frontiers without having raised any special problems.

The juridical aspect is covered in a note on the evolution of international law on transport matters.

These few examples suffice to show the importance of the work and also how the objectives of the authors have been attained. We have before us as it were a synthesis of the international creations of the railways accompanied by a programme of the methods to be taken in hand to promote the influence of railways in a unified Europe.

E. M.

[**385** (05 (43)]

Archiv für Eisenbahnwesen, 67th year, volume No. 1 (completed on the 3rd April 1957). Review published by the Central Administration of the German Federal Railways. — One volume (6 3/4 × 10 1/4 inches) of 132 pages. — 1957, Springer-Verlag, Berlin-Göttingen-Heidelberg, Berlin W. 35 (West Berlin), Reichpietschufer 20. (Price of the annual subscription : DM 66,—.)

This review is well known throughout the railway world. Begun in 1878, it is considered as a model of such reviews in Germany and is also widely bought and appreciated in other countries.

It therefore needs no introduction to our readers. We will merely say that after an eclipse of fifteen years, it has reappeared with this first new number in 1957.

As formerly, the volume includes a scientific portion and an administrative one.

The author of the first note gives the history of the " Archives ". This history is closely bound up with that of the German railways.

A second note deals with the administrative organisation and functioning of the railways which do not belong to the Bundesbahn.

A third note studies the development of the Netherlands Railways year by year from 1942 to 1955.

The various communications include an analysis of the operating results of the German Federal Railways in 1956, and of the Austrian Federal Railways and French Railways in 1955, information regarding the research work undertaken concerning the comparison of operating results, an appreciation of the treaty made between the German Federal Republic, the French Republic and the Grand Duchy of Luxemburg regarding the canalization of the Moselle. A few bibliographical notes complete the text.

The administrative portion gives the text of various laws with commentaries : the general Railway law of the 29-3-1951, the law on the Federal Railways of 13-12-1951 and 14-7-1953, the Administrative Decree of the German Federal Railways of the 24-3-1953 and two laws affecting the railways of Baden-Würtemberg and Schleswig-Holstein.

E. M.

[**656** .23]

RIDARELLI (G.). — **Molto o Polco? (Tariffe di ieri e di oggi).** — Quaderni delle Ferrovie Italiane dello Stato. 1957. *(Much or little? (Rates of yesterday and today).* Notes of the Italian State Railways. — One brochure (4 3/4 × 8 1/4 inches) of 96 pages, copiously illustrated. — 1957, published by the General Management of the Italian State Railways, Rome.

The question asked in the title is that discussed at the end of the work. After giving the history of railway rates in Italy, the author will state whether present day rates are to be considered high or moderate compared with other costs.

Originally, the rates, which were very simple and only concerned a short line, closely approximated the cost of transport by diligence.

As time went on, with the creation of new lines and increase in the distances, it became necessary to have a special rates structure based on technico-economic considerations.

It was not till 1848 when the Turin-Genoa line was opened to traffic that the rates were based on the passenger-kilometre and three classes were introduced.

The rates however differed from Company

to Company and often from line to line. The author reports how the political unification of the country facilitated the first important reform, the adoption of a general ordinary rates structure by the three great companies, which only took place in 1885.

Another great innovation was the introduction of differential rates. This only took place with difficulty and did not become general all at once. Finally, the charges diagram underwent several modifications.

As the years went on, the dramatic events which shook the country and the various economic upheavals had profound repercussions on the rates structure and the general level of rates.

After the second world war, several reforms were still necessary and the last to be introduced was reducing the number of classes from three to two. This obliged the Administration to make a new rearrangement in the rates, the most recent, which dates from July 1956.

Taking as his basis definite numerical data, the author does not hesitate to reply in the affirmative to the question of deciding whether a journey in Italy is still good value according to economic data.

Amongst the figures quoted, there is a table showing the various partial variations compared with 1938 prices on the one hand and 1939 prices on the other. Another table gives the coefficients of the increase in prices since 1885.

At the end of the volume there is a summary of all the rates applying to passenger transport in various periods since 1885.

Reading this little volume, which is particularly well illustrated, we appreciate once again how much transport, and in particular railway transport, helps the economic life of a nation. The author points out, moreover that Italian managerial policy has always had political and social ends in view which the undertaking must meet, and the great variation in the rates tends to satisfy the most varied requirements of the public.

E. M.

[**621** .3 (03]

PIRAUX (H.), Ingénieur-électricien, Chargé de cours à l'Enseignement Technique. — **Dictionnaire Anglais-Français des termes relatifs à l'Electrotechnique, l'Electronique et aux applications connexes** *(English-French Dictionary of terms used in connection with Electrotechnics, Electronics and similar applications)*. — Third edition revised and completed. — One volume (6 1/4 × 9 1/2 inches) of 308 pages. — 1956, Paris (V^e), Editions Eyrolles, 61, Boulevard Saint-Germain. (Price : 1 850 French francs).

This work preceded the French-English Dictionary the publication of which was announced in our September 1957 *Bulletin*. The fact that it has already reached a third edition bears witness to its success.

The techniques covered are rapidly spreading in many and varied fields. There are many French speaking students who wish to study English works on the subject. Their work will be greatly facilitated by these two dictionaries.

This one contains a translation of more than 20 000 words or expressions. Devoted in principle to electrotechnics, it naturally includes electronics, and amongst the subsidiary applications we may mention optics, acoustics, nuclear physics, radar, radio and television. This will give some idea of the abundance of the material covered.

During a period of expansion, terminology has not to be too definite. Various special organisations have undertaken the task of fixing the function of the new equipment and devices and standardising the names. Since the author was inspired by their labours, we are certain of finding the standard terminology in his dictionary.

The last pages contain a great number of Anglo-Saxon units of measurement with their value in metric units. This makes it unnecessary to the reader to consult other sources when he has to convert one to the other.

E. M.

[**656** .212 .5]

Rangiertechnik, Hefte 13, 14, 15, 16. (*Technical Railway Review.* Special issues 2, 5, 6 and 7.) *(Marshalling Technique.* Brochures Nos. 13, 14, 15 and 16*).* — Special numbers of the *Eisenbahntechnische Rundschau* : No. 2 (Dec. 1953), No. 5 (Dec. 1954), No. 6 (Dec. 1955) and No.7 (Dec. 1956). — Published by the Special Commission of Marshalling Technique of the German Federal Railways. — 4 issues (7 7/8 × 11 1/2 inches), respectively of 120, 94, 84 and 62 pages, with numerous illustrations. — Publishers : Carl Röhrig-Verlag HG, Darmstadt, Holzhofallee 33 *a.* (Price : DM 18.—, 16.—, 16.— and 10.—.)

Marshalling yards are extremely important to the railway because of the high cost of the installations and labour they involve, as well as their influence on the rapid turn round of the stock.

The German Federal Railways have set up a special commission to study this question of marshalling. The results of its work are published in special issues of the *Eisenbahn Technische Rundschau* which appear each year. The four last issues bear the numbers 13, 14, 15 and 16 and relate to the years 1953 to 1956.

Each issue starts with a report on the activities of the commission during the previous year (information extracted from the research archives) and then gives a study on marshalling technique.

The report mentions the problems which have been studied and on which a final or provisional decision has been reached.

For example, we may mention some of the problems examined during the year 1952 *(Issue No. 13 of December 1953)* :

Complete or part storing up of the marshalling itineraries;

Distant control of the shunting locomotives;

Influence of roller bearing axle boxes on the speed of shunted wagons;

Preliminary inspection during shunting;

Lighting by overhead floodlights in marshalling yards;

Method of determining the direct costs of marshalling yards;

The transmission of marshalling notes;

The use of radio in marshalling operations;

Construction and layout of points facilitating the descent of the wagons;

Automatically operated hump brakes;

Principles of remote control of lateral track brakes.

Certain of these problems are related to some of those extensively studied which take up most of *Issue No. 13.* For example, we find articles on :

The inspection of wagons and weighing them whilst pushing them up the hump and running down it;

The history of the lighting of marshalling yards;

Calculations relative to track brakes.

We shall have practically covered the series by mentioning in addition :

Taking the climate into account in evaluating the resistance to running in calculating the dynamics of running down the sidings;

Output limits in the case of complete automatisation of the shunting zone;

Simplification of making up the trains.

Shunting and all its attendant problems seems to be the main preoccupation. In *Issue No. 14,* we find another study on the mechanisation of brake shoes.

The question of the evaluation of costs is dealt with in two articles devoted to marshalling yards and the intermediate yards.

As regards the general conception of the installations, there is a study dealing with ladder tracks and a discussion on the advantages and drawbacks of a single or double access line to the hump *(Issue No. 15).*

In *Issue No. 16,* there is a discussion on yards with one approach road and those with two, one at opposite ends. Shunting is dealt with again in a note on developments in measuring the running resistance of wagons and a description of the Brs control equipment at Gremberg yard. This latter installation based on new principles results amongst other things in a great reduction in the number of shifts required.

These brief notes will suffice to show that we have here some scientific literature which can usefully be consulted by all those concerned with the equipment and operation of marshalling yards. E. M.

M. WEISSENBRUCH & Co. Ltd.
Printer to the King
(Manag. Dir. : P. de Weissenbruch,
238, chaussée de Vleurgat, XL)
Edit. responsible : P. Ghilain

PRINTED IN BELGIUM